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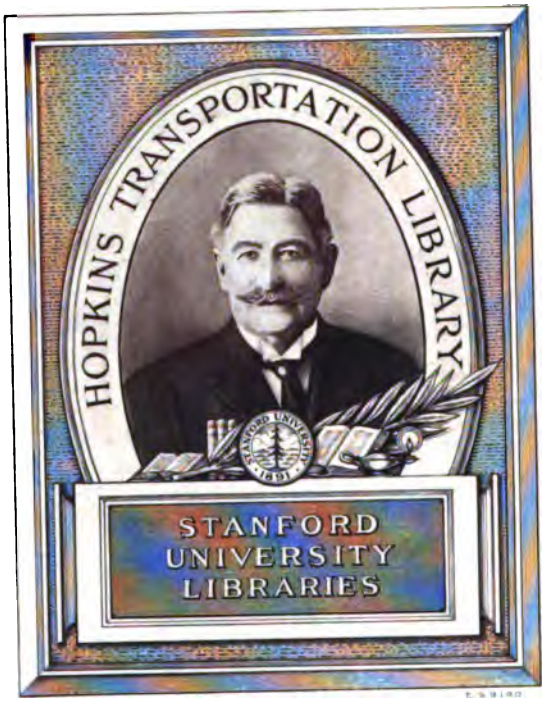

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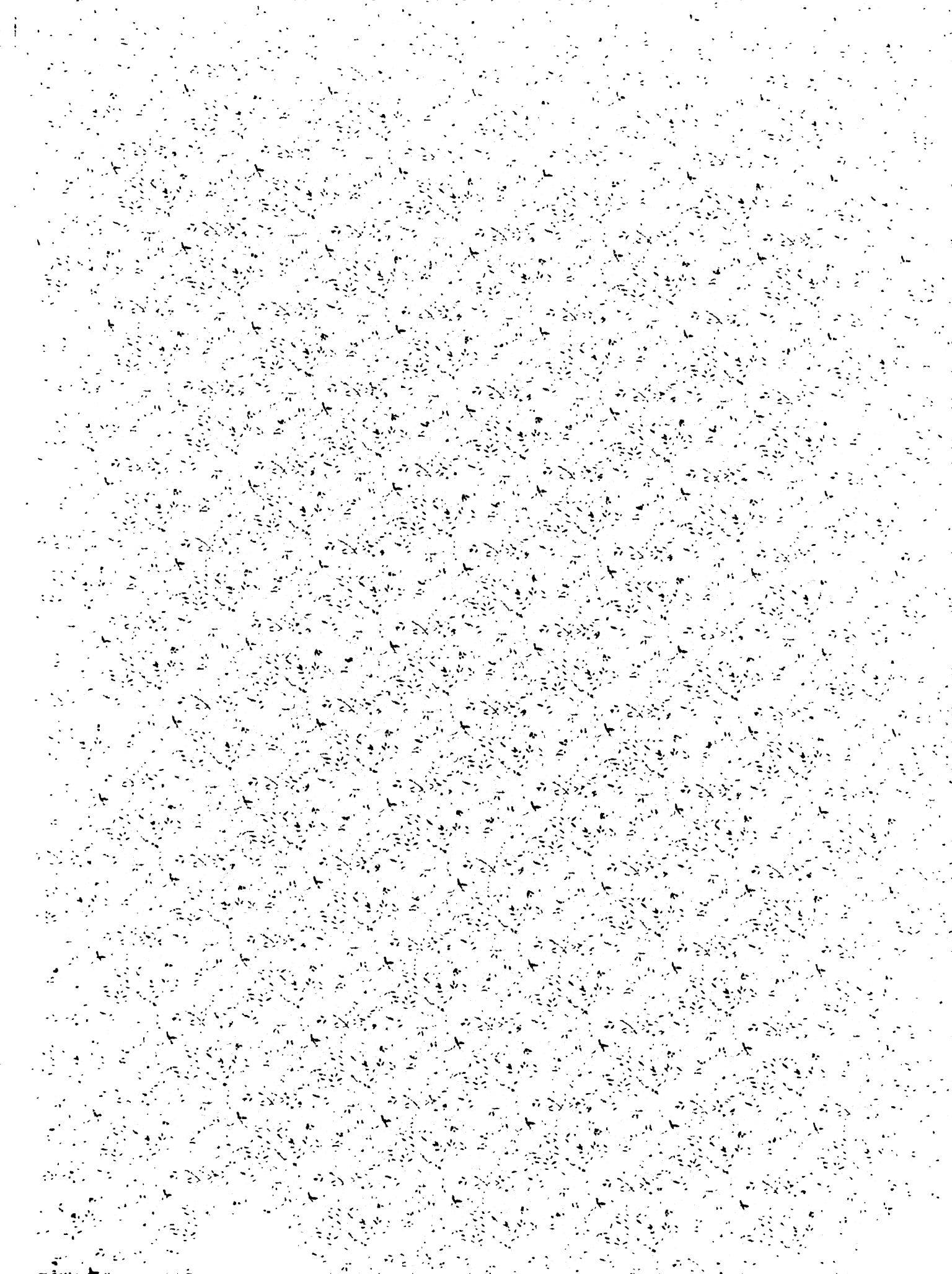

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VOL. IX 1901

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Norsworthy, Leonard D.

O'Brien, Edward C.
O'Connor, Bernard F.
Oliver, Anthony J.
Orcutt, Calvin B.
Ostheimer, Alfred J.

Paine, Cecil E.
Paschwitz, Hubert Von Rebeur.
Pattison, Hugh.
Patton, John B.
Paul, Lawrence T.
Perkins, Charles T.
Perry, Thomas.
Peterson, Otto L.
Petre, Axel E.
Pierce, Walter L.
Pollock, Walter B.
Pomeroy, L. R.
Post, Walter A.
Potter, Edward E.
Powell, Joseph W.
Powelson, Walfrid Van Ness.
Powers, Francis W.
Prout, H. G.
Pusey, Howard.

Quayle, George L.

Randebrock, Carl H.
Randle, William G..
Raven, Anton A.
Raymond, Charles H.
Raymond, James R.
Rea, Samuel.
Reed, Henry S.
Roach, Stephen W.
Robert, William P.
Roberts, Thomas G.
Robinson, Frank B.
Robinson, Richard H. M.
Rockwell, Charles H.
Rogers, Walter H.
Ross, Charles E.
Rossow, Ernest F.
Rowen, John H.
Rozycki, S. L. O.

Sabin, Alvah H.
Sampson, Wm. T.
Satterlee, Herbert L.
Schock, Edson B.
Schoeffel, George J.
Schooler, Clifford V.
Schouler, John.
Schreidt, Frank.
Schreiner, Watson C.
Scott, Henry T.
Sebree, Uriel.
Shackford, John W.
Shackford, William G.
Slazenger, Frank L.
Sloan, Horace T.
Smith, Alexander R.
Smith, Alfred G.
Smith, Godfrey L.
Smith, Stuart F.
Smith, Townsend J.
Sprague, F. J.
Stanton, Samuel W.
Starke, Frederick C.
Stephens, Wm. P.
Stilz, Harry B.
Swanton, Henry.

Taft, Harrison S.
Tams, James F.
Thayer, Charles H.
Thayer, Horace H.
Thresher, Alfred A.
Toch, Henry M.
Todt, Edward G.
Townsend, George R.
Trigg, William R.
Tromp, Theodore H. A.
Tuthill, Benjamin C.

Vandertoorn, Willen H.
Vanderbilt, Aaron.
Van Duzer, Louis S.
Veit, Richard C.
Vreeland, Charles E.

Wales, Charles M.
Walsh, John E.
Ward, Aaron.
Waters, George H.
Waters, J. W. Lowry.
Watson, Thomas A.
Weinz, Charles J.

Welles, Roger, Jr.
Wells, Benjamin W., Jr.
West, Charles C.
West, Gordon B.
Wheater, Frank R.
Wheeler, Charles E.
White, William P.
Whitney, Charles W.
Whiton, Chauncey G.
Wierdsma, John R.
Wilde, George F. F.
Willard, Charles F.
Williams, Henry.
Wilson, George P.
Wise, William C.
Witt, Oscar F.
Wolvin, August B.
Wood, Howard.
Wood, Wm. R. C.
Woods, Allen D.
Wright, Robert K.
Wright, William L.
Wyman, Wm. I.

Zerega, Theodore C.

JUNIORS (82).

Bacon, Henry D.
Ballord, Webb R.
Bean, Walter R.
Baum, Arthur E.
Bimson, James A.
Binley, Wm., Jr.
Bowen, Almon F.
Boyd, Stephen B.
Bretteville, Alexander de.
Briner, Emil A.
Brooks, Louis C.

Caswell, W. Benson.
Chase, Arthur W.
Christophersen, Herbert H.
Clark, Dean.
Clary, Joseph.
Crain, John J.
Crawford, James A.
Crosby, Ralph W.
Cutler, Edward O.

Damon, Edward O.
Davis, Matthews E.
Donnelly, Patrick J.
Dougherty, Hugh.
Ellis, Walter L.
Evans, John L.

Fernald, Benjamin G.
Fisher, Lee D.
Frieble, John F.

Gardner, J. Howland.
Godbold, Charles H., Jr.
Grieshaber, Hugo E.
Grogan, Harley H.

Haight, Robert S.
Hall, Francis G., Jr.
Herbert, Jerome P.
Hibbard, Harry L.
Hunnewell, Fred. A.
Hunt, Herman.
Hussey, Wm. F.

Jenkins, Sanford C.
Johnson, Eads.
Jonson, Herman.

Kimball, Walter E.
Kreer, John G.

Lee, Harold.
Leitch, Albert C.
Loomis, Allen.
Lord, Robert W.

Mager, Edwin E.
McComb, Samuel D.
McCormack, James W.
Meeker, H. L., Jr.
Mitchell, Leland R.
Moody, George B.

Newell, William S.
Newton, Wm. B.

Overmon, Austin E.

Page, Charles B.
Paulding, Charles P.
Pierce, Edward E.

Read, Walter A.

Salmond, James J.
Schoonmaker, Frederick.
Setchell, Horace E.
Sheridan, Richard B.
Stuart, Donald.
Swasey, Albert L.

Terry, Arthur L.
Trask, Edward P.
Tuck, Edward G.

Wakeman, Samuel W.
Whitaker, Morris M.
White, Warren A.
Whitehouse, Frederick C.
Whittelsey, Henry N.
Wight, Roger W.
Willis, William.
Winterburn, Charles.
Wright, Henry C., Jr.

Yeomans, Clifton.
Young, William J. J.

ARTICLES OF INCORPORATION.

SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS.

STATE OF NEW YORK,
CITY AND COUNTY OF NEW YORK: } *ss:*

We, the subscribers, WM. H. WEBB, CHAS. H. CRAMP, GEORGE E. WEED, H. TAYLOR GAUSE, WILLIAM T. SAMPSON, HORACE SEE, FRANK L. FERNALD, FRANCIS T. BOWLES, WASHINGTON L. CAPPS, EDWIN D. MORGAN, GEORGE W. QUINTARD, HARRINGTON PUTNAM, and JACOB W. MILLER, being persons of full age and citizens of the United States, of whom a majority—namely, William H. Webb, George E. Weed, Horace See, Edwin D. Morgan, George W. Quintard, Harrington Putnam, Frank L. Fernald, and Jacob W. Miller—are citizens of and residents of and within this State, desiring to associate ourselves for scientific purposes under, and pursuant to, an Act of the State of New York providing for the incorporation of benevolent, charitable, scientific, and missionary societies, passed April 12, 1848, and the several acts amending or supplementing the same, do hereby, in accordance with the requirements thereof, certify as follows:—

First. The name or title by which the Society shall be known in law is THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS.

Second. The particular business and objects of such Society are the promotion of practical and scientific knowledge in the arts of shipbuilding and marine engineering and the allied professions, and in furtherance of this object, to hold meetings for social intercourse among its members, and the reading and discussion of professional papers, and to circulate by means of publication the knowledge thus obtained.

Third. The number of directors, trustees, or managers to manage the Society shall be seven, and shall consist of a President, a Secretary, and five Members of Council.

Fourth. The names of the trustees, directors, or managers of the Society for the first year of its existence are: President, Clement A. Griscom; Secretary, Washington L. Capps; Members of Council, Francis T. Bowles, H. Taylor Gause, Chas. H. Loring, Lewis Nixon, Harrington Putnam.

Fifth. The business of the Society is to be conducted, and its place of business and principal office is to be located, in the City and County of New York.

IN WITNESS WHEREOF we have made, signed, and acknowledged this certificate, this 28th day of April, 1893.

WILLIAM H. WEBB.
CHAS. H. CRAMP.
H. T. GAUSE.
GEORGE E. WEED.
W. T. SAMPSON.
HORACE SEE.
F. L. FERNALD.
FRANCIS T. BOWLES.
W. L. CAPPS.
E. D. MORGAN.
GEORGE W. QUINTARD.
HARRINGTON PUTNAM.
J. W. MILLER.

CITY AND COUNTY OF NEW YORK, *ss:*

On this 28th day of April, 1893, before me personally appeared William H. Webb, Charles H. Cramp, H. Taylor Gause, George E. Weed, William T. Sampson, Horace See, Frank L. Fernald, Francis T. Bowles, Washington L. Capps, and Edwin D. Morgan, to me known and known to me to be the persons described in and who executed the foregoing certificate, and severally acknowledged to me that they executed the same.

JAMES FORRESTER,
Notary Public, Kings Co., Cert. N. Y. Co.

CITY AND COUNTY OF NEW YORK, *ss:*

On this 1st day of May, 1893, before me personally appeared George W. Quintard and Harrington Putnam, to me known and known to me to be the individuals described in and who executed the foregoing certificate, and they severally acknowledged to me that they executed the same.

JAMES FORRESTER,
Notary Public, Kings Co., Cert. N. Y. Co.

CITY AND COUNTY OF NEW YORK, *ss:*

On this 9th day of May, 1893, before me personally appeared Jacob W. Miller, to me known and known to me to be one of the individuals described in and who executed the foregoing certificate, and he duly acknowledged to me that he executed the same.

JAMES FORRESTER,
Notary Public, Kings Co., Cert. N. Y. Co.

(ENDORSED.)

Upon reading the within Certificate for the Incorporation of the Society of Naval Architects and Marine Engineers, I hereby approve and consent to the incorporation thereof and the within certificate and filing thereof, and direct that the same be filed in the office of the clerk of the City and County of New York.

Dated New York, May 10, 1893.

EDWD. PATTERSON,
Justice of the Supreme Court in the State of New York in and for the City and County of New York.

CONSTITUTION AND BY-LAWS OF THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS.

ARTICLE I.

Name and Object.

1. The name of this Association shall be "THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS."

2. Its object shall be the promotion of the art of shipbuilding, commercial and naval.

3. In furtherance of this object, annual meetings shall be held for the reading and discussion of appropriate papers and interchange of professional ideas, thus making it possible to combine the results of experience and research on the part of shipbuilders, marine engineers, naval officers, yachtsmen, and those skilled in producing the material from which ships are built and equipped.

ARTICLE II.

Membership.

1. The Society shall consist of Members, Associates, Juniors, Honorary Members, and Honorary Associates.

2. *Members.*—(1) The class of Members shall consist exclusively of Naval Architects, Marine and Mechanical Engineers, including Professors of Naval Architecture or Mechanical Engineering in colleges of established reputation.

(2) A candidate for this class must be not less than twenty-five years of age and comply with the following regulations:—He shall submit to the Council a statement showing that he has been engaged in the practice of his profession, in a responsible capacity, for at least three years, and setting forth his grounds upon which he bases his claim to membership. This statement shall be signed by three members, who shall certify to their personal knowledge of the candidate and approval of his statement.

(3) In the case of persons not American citizens, the signatures of three members shall be required in confirmation of their personal knowledge of the candidate's scientific attainments.

(4) If three-fourths of the members of the Council present are in favor of the admission of the candidate, his name shall be submitted to the members of the Society at the next meeting, the voting to be by ballot, should a ballot be demanded.

3. *Associates.*—(1) The class of Associates shall consist of all persons who, by profession, occupation, or scientific attainments, are qualified to discuss the qualities of a ship.

(2) Candidates for this class shall submit to the Council a written statement of their qualifications for membership. If considered by three-fourths of the Council present duly qualified for associate membership, their names shall be submitted to the Society at its next meeting, to be voted upon by the Members and Associates; voting to be by ballot, should a ballot be demanded.

4. The proportion of votes for deciding the election of Members and Associates shall be at least four-fifths of the number recorded.

5. *Juniors.*—(1) The class of Juniors shall consist of graduates of technical schools of established reputation, or persons who have not less than two years' practical experience in marine engine works or shipyards.

(2) Candidates must be at least eighteen years of age and certify their intention to continue in the profession and become naval architects or marine engineers.

(3) Juniors shall be eligible for transfer to the class of members after fulfilling the necessary conditions.

(4) The admission of Juniors shall be by a favorable vote of three-fourths of the members of the Council present.

(5) Juniors shall have no voice in the government of the Society or admission of members.

6. *Honorary Members and Honorary Associates.*—The Council may elect Honorary Members and Honorary Associates, the total number not to exceed twenty-five. They shall be persons of acknowledged eminence in their profession upon whom the Council may see fit to confer an honorary distinction.

Article III.

Dues.

1. The entrance fees, payable on admission to the Society, shall be as follows:—

Members and Associates, five dollars; Juniors, three dollars; Honorary Members and Honorary Associates, no fees.

2. The annual dues shall be as follows:—

Members and Associates, five dollars; Juniors, three dollars; Honorary Members and Honorary Associates, no dues.

3. A member transferred from one grade to another shall pay the difference between the entrance fees of the two grades, and his annual dues shall be those of the grade to which transferred.

4. The annual contributions shall be payable in advance on the first day of January. The Secretary shall notify each member of the amount due for the ensuing year at the time of giving notice for the annual meeting.

5. Members and Associates can compound for all future dues and become Life Members and Life Associates, by making a single payment of two hundred dollars and signing an agreement to conform to any future amendments to the Constitution and By-Laws.

6. Members are entitled to no return of fees upon severing their connection with the Society.

7. Any member whose dues are more than three months in arrears shall be notified by the Secretary. Should his dues become six months in arrears he shall be again notified by the Secretary and his rights as a member suspended. Should his dues become one year in arrears, the delinquent member shall forfeit his membership in the Society unless the Council may deem it expedient to extend the time of payment.

8. The Council may, in its discretion, temporarily suspend the annual payment of dues by any member whose circumstances have become such as to make such payment impossible, and may, under similar circumstances, remit the whole or a part of dues in arrears.

9. No name shall be entered on the rolls as Member, Associate or Junior, nor shall the privilege of membership be enjoyed until all dues shall have been paid; if the payment be delayed for more than six months from the date of election the same shall be void unless the Council otherwise direct.

10. (1) Should the expulsion of any member be judged expedient by five or more members, they must draw up and sign a proposal requesting such expulsion, delivering the same to the Secretary, to be laid before the Council.

(2) If the Council do not find reason to concur in the proposal, no entry thereof shall appear in the minutes, nor shall any public discussion thereon be permitted.

(3) If, however, the Council find the charges contained in the proposal for expulsion substantiated, the accused member shall be notified and given an opportunity to resign. If he avails himself of this privilege, no entry shall be made on the minutes nor public discussion of the case permitted; but if he declines to resign and offers no satisfactory explanation of the charge, the whole case shall be submitted to a special meeting of the Society.

(4) If two-thirds of the members at this special meeting (provided there be not less than twenty present) vote for expulsion, the chairman of the meeting shall cause the accused to be expelled from the Society, and direct the Secretary to notify the accused of this action.

ARTICLE IV.

Officers.

1. The officers of the Society shall consist of a President, Vice-Presidents, Members of Council, a Secretary and Treasurer.

2. Both Members and Associates are eligible for the offices of President, Vice-

President, and Members of Council, but at least three-fourths of the Council shall be composed of Members.

3. The officers for the ensuing year shall be elected at the annual general meeting.

4. A vacancy in the office of President shall be filled by the Senior Vice-President until the next annual meeting, when the vacancy shall be filled by election in the ordinary way.

5. The President and Vice-President and all those who have held those offices shall be *ex officio* members of Council, and entitled to sit and vote with that body as long as they continue their connection with the Society.

6. The Council may hold meetings, subject to the call of the President, as often as the interests of the Society may demand.

7. At all meetings of Council, five members shall constitute a quorum.

8. The Secretary and Treasurer shall be elected annually by the Council, but may be removed at any time by a majority vote of the Council after due notice has been given.

9. The Secretary must be a member of the Society.

ARTICLE V.

Management.

1. The President shall have general supervision over the affairs of the Society, appoint special committees, and preside at the annual general meetings. He shall be *ex officio* member of all committees.

(2) In the absence of the President, the Vice-Presidents, in the order of seniority, shall preside and perform all the duties of the President.

2. The direct management of the Society shall be vested in an Executive Committee composed of five members of Council elected annually by the Council, the President and Secretary of the Society being *ex officio* members of the Committee. At least three of the five elective members of the Committee shall be Members of the Society.

3. The Executive Committee shall manage the affairs of the Society in conformity with the laws under which it is incorporated, and the provisions of the Constitution. It shall direct the investment and care of the funds of the Society; make appropriations for specific purposes; arrange for the reading and publication of professional papers under such regulations as the Council may, from time to time, prescribe; take measures to advance the interests of the Society, and generally direct its affairs.

4. The Executive Committee shall make an annual report to the Society, transmitting the report of the Secretary and Treasurer and of any special committee which may have been ordered.

5. The Secretary shall be the Executive Officer of the Society under the immediate direction of the President and Executive Committee.

(2) He shall prepare the business for the annual meetings and record the proceedings thereof.

(3) He shall be responsible for all expenditures and certify the accuracy of all bills or vouchers upon which money has been paid, and he shall conduct the correspondence of the Society and keep full records of the same.

6. The Treasurer shall see that all money due the Society is collected and carefully invested in such manner as the Executive Committee may direct. If considered advisable by the Council, the duties of Treasurer may be performed by the Secretary.

7. The accounts of the Secretary and Treasurer shall be audited annually by a special committee of three members of the Council appointed by the President of the Society.

ARTICLE VI.

Meetings.

1. There shall be at least one annual general meeting of the Society for the reading and discussion of professional papers, election of officers for the ensuing year, and transaction of such other business as may be brought before it. The time and location of this meeting shall be determined by the Council at least three months prior to the date fixed.

2. Special meetings may be called by the Executive Committee at the request of twenty members, which request shall state the purpose of the meeting. The call for such meetings shall be issued ten days in advance, and shall state the purpose thereof. At these meetings, thirty members shall constitute a quorum.

3. The Society may adopt from time to time such rules as it may think proper for the order of business at its meetings.

4. Special meetings of the Executive Committee may be held at any time subject to the call of the Chairman; and four members shall constitute a quorum for the transaction of any business that may be properly brought before the Committee.

ARTICLE VII.

Amendments.

1. Proposed amendments to this Constitution must be reduced to writing and signed by not less than ten members. They shall be forwarded to the Secretary at least ten days before the annual general meeting, and shall be immediately forwarded to the Council for its consideration. If a majority of the Council approve the proposed amendment, it shall be presented to the Society at the next ensuing general meeting for discussion; if approved by two-thirds of the members present, voting by ballot, if a ballot be demanded, it shall be adopted.

INTRODUCTORY PROCEEDINGS.

NINTH ANNUAL MEETING OF THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS.

NOVEMBER 14TH AND 15TH, 1901.

The Ninth General Meeting of the Society of Naval Architects and Marine Engineers was held November 14 and 15, 1901, in the Auditorium of No. 12 West 31st Street, New York City. The meeting was called to order by the Secretary at 10.30 A. M., on Thursday, November 14th, and on account of the unavoidable absence of the President, it was stated that nominations for a temporary presiding officer were in order.

On motion of Mr. E. Platt Stratton, Mr. Stevenson Taylor was nominated and unanimously elected chairman in the absence of the President.

Mr. Stevenson Taylor then took the Chair.

THE CHAIRMAN:—Gentlemen, you will please come to order and listen to the report of the Secretary-Treasurer.

THE SECRETARY:—By resolution of the Council I announce to you that the report of the Secretary-Treasurer, so far as it relates to the general affairs of the Society, has been adopted, and will be presented to the Society as the report of the Council, the financial statement therein contained having been duly audited by a Committee of Council; the applications of candidates for membership have been fully considered by the Council, and its recommendations thereon will be submitted after presenting the report of the Secretary-Treasurer.

REPORT OF SECRETARY-TREASURER.

SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS,
12 WEST 31ST STREET, NEW YORK, *November 13, 1901.*

To the Council,
Society of Naval Architects and Marine Engineers,
New York.

GENTLEMEN:—I have the honor to submit the following report concerning the affairs of the Society for the year ending November 13, 1901:—

The membership of the Society at the present date is as follows:—

Style of Members.	At last report Nov. 15, 1900.	Elected 1900.	1900. Transferred to.	1900. Transferred from.	Deaths 1900.	Resignations 1901.	Dropped 1900.	Membership Nov. 13, 1901.
Members, ...	368	31	8	—	4	2	2	399
Associates, ...	244	45	2	6	2	3	1	279
Juniors, ...	49	25	—	4	—	1	—	69
Life Members, ...	5	—	—	—	—	—	—	5
Life Associates, ...	3	—	—	—	—	—	—	3
Totals, ...	669	101	10	10	6	6	3	755

The deaths and resignations that have occurred during the year are:—

Deaths (6).

Members (4)—

Alexander Henderson.
James S. Doran.
N. R. Benson.
H. W. Cramp.

Associates (2)—

Wm. E. Fitzgerald.
Rear-Admiral F. M. Bunce, U. S. N.

Resignations (6).

Members (2)—

David Joy.
Lieut. R. K. Crank, U. S. N.

Associates (3)—

Rear-Admiral N. H. Farquhar, U. S. N.
Isaac C. Greene.
Oswald Sanderson.

Juniors (1)—

D. W. Blake.

The following delinquencies exist in the payment of dues:—

From and including year ending December 31, 1899:—

Members and Associates	6
Juniors	1

From and including year ending December 31, 1900:—

Members and Associates	22
Juniors	2

For year ending December 31, 1901:—

Members and Associates	78
Juniors	10
Total	119

There is due from the above 119 members $566.

During the year, there have been sold:—

Of Vol. 1	20 copies.
Of Vol. 2	21 copies.
Of Vol. 3	17 copies.
Of Vol. 4	13 copies.
Of Vol. 5	10 copies.
Of Vol. 6	6 copies.
Of Vol. 7	8 copies.
Of Vol. 8	84 copies.

and there are now on hand:—

Of Vol. 1	537 copies.
Of Vol. 2	237 copies.
Of Vol. 3	251 copies.
Of Vol. 4	216 copies.
Of Vol. 5	109 copies.
Of Vol. 6	141 copies.
Of Vol. 7	119 copies.
Of Vol. 8	276 copies.

NOTE:—The report of agent in Berlin, showing volumes on hand for year ending November 13, 1901, has not yet been received.

The following candidates have made application for membership:

Members	17
Associates	37
Juniors	12
Total	66

Advancement from Associate to Member	4
Advancement from Junior to Member	3
Advancement from Junior to Associate	1
Total applications for advancement	8

FINANCIAL

Receipts.

From November 14, 1900, to November 13, 1901.

Entrance fees and dues of new members for the year ending December 31, 1900:—		
Members and Associates (73)	$725 00	
Juniors (24)	144 00	$869 00
Dues for year ending December 31, 1901:—		
Members and Associates (595)	$2,975 00	
Juniors (58)	178 00	
Differences by transfers in membership	10 00	3,159 00
Delinquent dues for the year ending December 31, 1900:—		
Members and Associates (54)	$270 00	
Juniors (3)	9 00	
Differences by transfers in membership	4 00	283 00
Delinquent dues for year ending December 31, 1899:—		
Members and Associates (4)		20 00
Dues in advance for the year ending December 31, 1902:—		
Members and Associates (1)	$5 00	
Juniors (1)	3 00	8 00
Entrance fees and dues in advance from applicants for membership for year 1901:—		
Members and Associates (5)	$45 00	
Juniors (1)	6 00	51 00
Sales:—		
Papers		488 05
Volume 1, to Members (18)		56 00
Volume 2, to Members (18)	$56 00	
Subscriber (1)	6 00	62 00
Volume 3, to Members (17)		53 00
Volume 4, to Members (12)	$38 00	
Subscriber (1)	6 00	44 00
Volume 5, to Members (9)	$45 00	
Subscribers (1)	6 00	51 00
Volume 6, to Members (2)	$10 00	
Subscribers (3)	18 00	28 00
Volume 7, to Subscribers (7)		30 00
Volume 8, to Members (2)	$10 00	
Subscribers (76)	454 00	464 00
Receipts from foreign agents		36 00
Interest on deposit and bond		324 20
Exchange		13 66
Receipts from banquet 1900, and tickets 1901		734 60
Miscellaneous receipts, price competition		100 00
Total receipts		$6,874 51
Balance on hand November 14, 1900		1,692 74
		$8,567 25

STATEMENT.

Expenditures.

From November 14, 1900, to November 13, 1901.

Publishing Volume 8, Transactions 1900, and extra copies of papers		$4,829 28
Expenses Eighth Annual Meeting in New York, November 15th and 16th, 1900		174 30
Sending out copies of volumes to members and subscribers		289 43
Rent of quarters at 12 West 31st street		60 00
Salary of clerk from November 1, 1900, to November 1, 1901		300 00
Salary of secretary-treasurer from November 1, 1900, to November 1, 1901		500 00
Office expenses, stationery, messenger service, postage, etc		188 21
Exchange		18 78
Storage		34 50
Miscellaneous printing		246 15
Printing notices, invitations, banquet tickets, etc., for Ninth Annual Meeting, 1901,		17 07
Expenses Eighth Annual Banquet, New York, November 16, 1900		671 10
Balance on hand November 13, 1901		$7,328 82
In bank	$1,236 43	
In drawer	2 00	
		1,238 43
		$8,567 25

Audited and found correct, November 13, 1901.

J. E. Denton.
Lewis Nixon.
Frank B. King.

Statement of the Resources and Liabilities, November 13, 1901.

RESOURCES.

Membership fees due from members after deducting doubtful accounts.	$466 00
Due from subscribers for volumes	69 00
Due from dealers for volumes	371 98
Publications on hand as per inventory	5,134 85
Office furniture	80 00
Cash in bank	1,236 43
Cash on hand	2 00
Investment in bond	5,102 38
Certificate of deposit	3,000 00
Total resources	$15,462 64

LIABILITIES.

None.

Society's present worth	$15,462 64

Respectfully submitted.

W. L. CAPPS,
Secretary-Treasurer.

The SECRETARY:—Mr. Chairman, the Council have examined the applications of the following candidates for membership, and recommend them to the Society for election to the classes indicated:—

Members (17).

Roland Allwork, Inspecting Engineer, Panama R.R. & S.S. Line, care of Panama R.R. Co., 24 State St., New York.

John Millington Blankenship, Asst. Supt. Merchants' and Miners' Transportation Co., Baltimore, Md.

Jacob Brauchli, Inspecting Draughtsman, Office of Superintending Engineer, Newport News S. B. & D. D. Co., Newport News, Va.

William Ledyard Cathcart, Adjunct Professor, Mechanical Engineering, Columbia University, New York.

William Brown Collison, First Class Draughtsman, C. & R. Dept., Navy Yard, New York.

B. B. Crowninshield, Naval Architect, 31 State St., Boston, Mass.

Robert Everett Hall, Secretary, N. Y. Steam Fitting Co., 137 Elm St., New York.

Urban Harvey, 2d Asst. Engineer, Revenue Cutter Service, Division Revenue Cutter Service, Treasury Dept., Washington, D. C.

Vicente L. Legarda, Naval Architect, Union Iron Works, San Francisco, Cal.

Philip Leventhal, Designing Engineer and Chief Draughtsman, Gardner & Cox, 1 Broadway, New York.

Patrick H. McGiehan, Draughtsman, C. & R. Dept., Navy Yard, New York, 502 Jefferson Ave., Brooklyn, N. Y.

Stewart H. Moore, U. S. Local Inspector of Boilers, 2 River St., Chicago, Ill.; 6801 Perry Ave., Chicago, Ill.

Yashnichi Sugitani, Ship Yard Manager, Mitsu Bishi Dock Yard & Engine Works, Nagasaki, Japan.

Paul Augustine Talbot, Chief Hull Draughtsman, Wm. R. Trigg Co., Richmond, Va.

Frank Van Vleck, Asst. Superintending Engineer, U. S. Army Transport Service, 36 New Montgomery St., San Francisco, Cal.

Harry Wheeler, Superintendent, Atlas S.S. Co., Pier 44, North River, New York.

Nagakato Yamamoto, Chief Draughtsman, Ship Dept., Mitsu Bishi Dock Yard & Engine Works, Nagasaki, Japan.

Promotion from Associate to Member (4).

Wm. G. Groesbeck, Asst. Naval Constructor, U. S. N., Wm. R. Trigg Co., Richmond, Va.

Joseph E. McDonald, Asst. Naval Constructor, U. S. N., Navy Yard, Mare Island, Cal.

Joseph B. Rowen, Asst. in Charge of Mold Loft, Wm. Cramp & Sons' S. & E. B. Co., Philadelphia, Penna.

Henry H. Schulze, Leading Draughtsman, in Charge of Merchant Work, Wm. R. Trigg Co., Richmond, Va.

Promotion from Junior to Member (3).

Thomas M. Cornbrooks, Chief Draughtsman, Hull Dept., Maryland Steel Co., Sparrow's Point, Md.

Walter S. Leland, Instructor in Naval Architecture, Mass. Inst. of Technology, Boston, Mass.

Alpheus Appleton Packard, in Charge of Ordering Department, N. Y. Shipbuilding Co., Camden, New Jersey.

Associates (37).

Herbert Lincoln Aldrich, Publisher and Manager Marine Engineering, 309 Broadway, New York.

Elliott Ivar Gamborg Andresen, Asst. Manager, Holzafel's Composition Co., 18 Broadway, New York.

George Harold Bates, Chief Engineer, Burlee Dry Dock Co., Port Richmond, S. I.; Box 40, Crawford, N. J.

Guy Aloysius Bisset, Asst. Naval Constructor, U. S. N., Navy Yard, Boston, Mass.

Harry Brown Bloomsburg, Draughtsman, Engine Dept., Newport News, S. B. & D. D. Co., Newport News, Va.

Walter William Clark, Engine Draughtsman, Neafie & Levy, S. & E. B. Co., Philadelphia, Penna.

Alexis Crane, N. Y. Manager and Special Expert Ship Equipment, Art Metal Construction Co., 346 Broadway, New York.

Frank H. Cross, Hull Draughtsman, Wm. R. Trigg Co., Richmond, Va.

Karl Erik Delin, Draughtsman, Hull Dept., Newport News S. B. & D. D. Co., Newport News, Va.

Thomas Merritt Dick, Ensign, U. S. N. (retired), Professor of Mechanical Engineering, A. & M. College, Raleigh, N. C.

Edmund Baker Edwards, Draughtsman, Fore River S. & E. Co., Quincy, Mass.

Harvey D. Goulder, Proctor in Admiralty, 816 Perry-Payne Building, Cleveland, Ohio.

James Ringold Guy, B. S., Hull Draughtsman, Wm. R. Trigg Co., Richmond, Va.

O. P. Hagstrom, Leading Draughtsman, Newport News S. B. & D. D. Co., Newport News, Va.

Bernard Hedstrom, Hull Draughtsman, Moran Brothers Co., Seattle, Wash., P. O. Box 873, Seattle, Wash.

Gabriel Hofgaard, Ship Draughtsman, Wm. Cramp & Sons' S. & E. B. Co., Philadelphia, Penna.

Philip Lamson, Draughtsman, Hull Dept., Newport News S. B. & D. D. Co., Newport News, Va.

Frank Provoost Larkin, Draughtsman, Engineering Dept., Newport News S. B. & D. D. Co., Newport News, Va.

Newbury D. Lawton, 40 Wall St., New York.

Curtis Darling Mabry, Hull and Calculating Draughtsman, Gardner & Cox, 1 Broadway, N. Y.; 226 West Lincoln Ave., Mt. Vernon, N. Y.

Lewis Albert Mason, Chief Hull Draughtsman, Burlee Dry Dock Co., Port Richmond, S. I.

Joseph McCreery, Senior Member, Joseph McCreery & Sons, Toledo, Ohio.

Wm. Cullen McGowan, Asst. Draughtsman, Wm. R. Trigg Co., Richmond, Va.

Charles C. Moss, Ship Draughtsman, Wm. R. Trigg Co., Richmond, Va.

Leonard Drake Norsworthy, Ship Draughtsman, Bath Iron Works, Bath, Maine; 146 Oak St., Bath, Maine.

Charles Thomas Perkins, Draughtsman, C. & R. Dept., Navy Yard, Portsmouth, N. H.

Harry Ball Stilz, Ship Draughtsman, Naval Constructor's Office, Bath Iron Works, Bath, Maine.

Horace Holden Thayer, Jr., Estimating Draughtsman, Harlan & Hollingsworth Co., Wilmington, Del.

Henry M. Toch, Manufacturer of Marine Cements and Ship Bottom Paints, 472 West Broadway, New York.

Wm. Robertson Trigg, President, Wm. R. Trigg Co. and V. P. Richmond Locomotive Works, Richmond, Va.

Charles J. Wemz, Superintendent Insulating Construction, H. W. Johns Mfg. Co., 100 William St., New York.

Charles Cameron West, Draughtsman, Hull Dept., Chicago Shipbuilding Co., Chicago, Ill.

Chauncey Gilbert Whiton, Treasurer and Agent, The New Bedford, Martha's Vineyard & Nantucket Steamboat Co., 7 Commercial St., New Bedford, Mass.

Charles Franklin Willard, Scientific Dept., Eastern Shipbuilding Co., New London, Conn.; P. O. Box 336, Groton, Conn.

Henry Williams, Asst. Naval Constructor, Navy Yard, New York.

Oscar Fredrik Witt, Hull Draughtsman, Newport News S. B. & D. D. Co., Newport News, Va.

Robert Kemp Wright, Master of Steam Yacht Nourmahal, New York Yacht Club, 44th St., New York.

Promotion from Junior to Associate (1).

Charles Haynes Hughes, in Charge of Torpedo Boats under Construction, Crescent Shipyard, Elizabeth, New Jersey.

Juniors (12).

Henry Douglas Bacon, Draughtsman, Hull Dept., N. Y. Shipbuilding Co., Camden, N. J.; 115 Evergreen Avenue, Woodbury, N. J.

Alexander de Bretteville, Draughtsman, Fulton Engineering and Shipbuilding Co., San Francisco, Cal.

Emil Amandus Briner, M. E., Neafie & Levy S. &. E. B. Co., 1917 Girard Avenue, Philadelphia, Penna.

James Alexander Crawford, Asst. Draughtsman, Naval Constructor's Office, Union Iron Works, San Francisco, Cal.; 414 19th St., Oakland, Cal

Hugh Dougherty, C. & R. Dept., Navy Yard, New York ; 74 Manhattan Ave., Jersey City Heights, N. J.

Charles Henry Godbold, Jr., Draughtsman, Union Iron Works, San Franscisco, Cal.; 2040 Clinton Avenue, Alameda, Cal.

Francis Granger Hall, Jr., Asst. Manager, Naval Electric Co., 95 Liberty St., N. Y.

John G. Kreer, with Purdy & Henderson, 539 La Salle St., Chicago, Ill.

George Barrell Moody, Government Draughtsman, Bath Iron Works, Bath, Maine.

Walter Augustine Reed, Asst. Draughtsman, C. & R. Dept., Navy Yard, New York.

Arthur Lindville Terry, Jr., Draughtsman, Hull Dept. N. Y. Shipbuilding Co., Camden, N. J.

Roger Willard Wight, Asst. Draughtsman, Bath Iron Works, Bath, Me.

THE CHAIRMAN :—What is your pleasure, gentlemen, in regard to these candidates for membership, associate membership and junior membership?

Mr. JOHN C. KAFER :—I move that the action of the Council be approved by the meeting.

(The motion was put and unanimously carried.)

THE CHAIRMAN :—The gentlemen whose names are on the printed list which is before you, and which has just been read over by the Secretary, are therefore declared duly elected to membership in the Society.

THE CHAIRMAN :—The time has now come when the election of officers for the ensuing year is usually held. I will ask the Secretary to read to you the recommendation of the Council.

THE SECRETARY :—The Council has recommended the re-election of the old officers and the promotion of Admiral Bowles from Member of Council to Vice-President, vice Admiral Bunce, deceased. The promotion of Admiral Bowles and the resignation of Mr. H. Taylor Gause leaves two vacancies in the Council, and the Council recommends that these vacancies be filled by the election of Naval Constructor D. W. Taylor, U. S. N., and Professor Ira N. Hollis, of Harvard University.

THE CHAIRMAN :—What is your pleasure, gentlemen, in regard to these recommendations?

MR. KAFER :—I move that they be approved by the Society.

(The motion was unanimously carried.)

THE CHAIRMAN :—The recommendation of the Council having been unanimously approved, the present officers of the Society will continue for the ensuing year, subject to the changes mentioned by the Secretary.

THE CHAIRMAN : On account of the unavoidable absence of our President, you will be deprived of the pleasure of hearing his annual address, which is always so interesting and full of valuable information concerning current shipbuilding progress.

You have been informed of the present condition of the Society through the report of the Secretary-Treasurer, and you will observe that although large gains in membership have been made the Society has had the misfortune to lose some of its most distinguished members, among them being one of our Vice-Presidents, the late Rear-Admiral Francis Marion Bunce, United States Navy, whose distinguished services to his country need no comment from me.

I shall not attempt to offer you a substitute for the President's address, as I know that it would be but a poor substitute at best, and keep you from the consideration of the interesting programme of papers prepared for us; but there is just one subject I wish to touch upon before taking up the formal papers.

Through the liberality of a member of Council a prize was offered for the best paper on "The Theoretical and Practical Methods of Balancing Marine Engines." A committee of the Council composed of Messrs. J. E. Denton, Stevenson Taylor, and Lewis Nixon was appointed to examine the papers submitted in competition, and to recommend the paper entitled to the prize. Two of the papers submitted were of such surpassing excellence that the Committee recommended to the Council that an additional prize be offered in order that the paper which did not win the first prize might have suitable recognition.

The Council unanimously approved this recommendation of the Committee on Awards, so that a prize has been awarded to the author of the prize competition paper having as its motto "Let truth be our guide," the first prize having been unanimously accorded to the author of the paper having as its motto "Labor omnia vincit."

The Society is to be congratulated upon having presented to it papers of such unusual merit as those to whose authors prizes have been awarded, and the Council has decided to confer on the Executive Committee discretionary authority to offer a prize or prizes for papers to be read at our next meeting, the subject of the prize to be determined hereafter, and the value not to exceed two hundred dollars.

We will now proceed with the regular programme of papers, the first on the list being one prepared by Mr. Frank E. Kirby, Member of Council, on the "Trial of Speed between the Steamers City of Erie and Tashmoo." Mr. Kirby wrote

me a letter a few days ago, saying it would be impossible for him to be here, and asking me to read his paper and give you all his kindest regards, and regrets that he could not be with you. If there is no objection, I will read it from the desk without asking anyone to take my place as Chairman.

The Chairman then read Mr. Kirby's paper :—

TRIAL OF SPEED BETWEEN THE STEAMERS CITY OF ERIE AND TASHMOO.

By Frank E. Kirby, Esq., Member of Council.

[Read at the ninth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 14 and 15, 1901.]

On June 4, 1901, a trial of speed took place on Lake Erie between the Paddle Steamers City of Erie and Tashmoo. The course was straight along the south shore of the lake, starting from a line off the water-works crib six miles outside of the breakwater at Cleveland, Ohio, to a line ten miles off Presq'ile Light-house, at Erie, Pa. The distance measured on the United States Coast Survey Chart, which is on a scale 1/400,000, is 94 statute miles. Computed from the latitude and longitude of the light-houses at Cleveland and Erie, and corrected for location of starting line, the distance is 94.31 miles. Measured on the chart issued by the Hydrographic Office of the Navy Department, the distance is 95.25 miles. The calculations in this paper are based on a distance of 94 miles. The nearest land to the course was at Fairport, 2 1/2 miles distant. The average depth of water along the course was 61.31 feet; minimum depth, 35 feet; maximum depth, 77 feet. Weather fine; sea smooth.

Lines of both steamers, draught of water, and trim on date of trial are shown on Plate No. 3.

The City of Erie is a passenger and deck-freight steamer, owned and operated by the Cleveland and Buffalo Transit Company; in daily service with the steamer City of Buffalo, between the ports of Cleveland and Buffalo. Distance between harbor entrances, 173 miles; running time, ten hours. Best service run, eight hours, thirty-two minutes.

The hull is built of mild steel, divided into ten compartments, by nine bulkheads, six of which are water-tight. The upper decks and houses are of wood, arranged in the usual style of American steamboats.. Number of state-rooms, 163. Number of regular passengers allowed by permit, 600; excursion permit, 2,700. This could be increased by providing additional equipment. Number of officers and crew, 108. Capacity for freight, 600 tons.

Engine, single, compound beam, driving Feathering paddle-wheels.

Boilers, six; cylindrical, return tubular. Pressure of steam, 130 pounds. Amount of coal used per round trip in forty-eight hours, 67½ tons; amount of coal used per round trip of twenty-four hours, 55 tons. Coal used on the run of June 4th was select bituminous lump from the Youghiogheny district of the Pittsburg Coal Company's "Eclipse Mine."

The Tashmoo is an exclusively passenger steamer, owned and operated by the White Star Line; in daily service on Detroit and St. Clair rivers, crossing Lake St. Clair. The round trip between Detroit and Port Huron, occupies eleven and one-half hours; twenty landings are made each way, and lying at Port Huron two hours. Most of the run is in shallow water, which includes the St. Clair ship canal and the narrow cuts in Lake St. Clair. She has three complete decks for passengers; ten parlors, café and buffet on main deck; smoking-room on upper deck. Number of passengers allowed by permit, 3,000. Number of officers and crew, 84.

Engines, inclined, triple expansion, driving Feathering paddle-wheels. Boilers, cylindrical, return tubular—two double ended, and three single ended. Steam pressure, 175 pounds. Consumption of coal per day, 27 tons. Coal used when running with City of Erie was bituminous, from the Jackson Hill Mine, Jackson County, O. While the usual service of the Tashmoo is in smooth water, she is designed, built and classed as a lake steamer, making occasional runs across the head of Lake Erie to Toledo, the southern terminus of the White Star Line. She has a heavy equipment of anchors, chains, and eight large life boats. Her peculiar cross-section was designed to allow the boilers to go under deck, thus leaving the main deck clear for passengers. The thin ends of the hull were cut off to reduce the weight, and allow turning a reverse curve of 180 degrees in narrow waters at two landing places on her regular route. The only novel feature in her design is the gear for handling the water ballast used for trimming athwartship, which is shown on Plate No. 4, and has proved of great value. Tanks are built on the guards, just forward of the wheel housings, having a capacity, when full, of 13 tons of water each. Time required to fill one tank, seven minutes, the discharge water from the air pump being used. Time required to discharge, 45 seconds. The valves are arranged that, in any position, there is always a full outlet from the air pump. A connection is fitted for filling the tanks by the fire pump when main engine is not in motion. One man easily handles the gear, and his services are only occasionally required, except at landings.

Particulars of the steamers and data of the speed trial on June 4th, as follows:—

	City of Erie.	*Tashmoo.*
Length over all, feet	324	308
on keel, feet	314	300
Beam, feet	44	37·5
Breadth over guards, feet	77.2	69
Depth, feet	18	13.5
Draught, forward, feet	9.75	8.12
aft, feet	10.83	8.46
mean, feet	10.29	8.29
Displacement, tons	2,233	1,224
Trial load, tons	250	60
Wetted surface, square feet	12,776	8,976
Prismatic coefficient	.603	.584
Engine, type	Compound, beam.	Triple, inclined.
Cylinder, H. P. diameter, inches	52	33
I. P. diameter, inches	—	51
L. P. diameter, inches	80	82
stroke, inches H. P.	96	72
stroke, inches L. P.	144	—
Air pump, diameter, inches	52	45
stroke, inches	45	24
Feed pump, independent	16 x 11 x 14	14 x 8 x 12
Fire pump	14 x 8 x 10	10 x 6 x 10
Boilers, cylindrical, number	6	5
diameter, feet	12.5	11.08
length, feet	11.75	3 of 11.33 2 of 22
furnaces, diameter, inches	46	42
grate length, feet	5·5	6
surface, square feet	252	294
heating surface, square feet	11,580	8,750
Draft system	Howdens.	Natural.
Steam pressure, pounds	130	175
Blowers, number	2	—
diameter, inches	66	—
discharge, inches	34 x 34	—
engine, inches	7 x 7	—
heater, diameter, inches	25	25
length, feet	10.17	9·5
tubes, diameter, inches	1½	1½
length, feet	8	7
surface, square feet	302	282
Wheels, diameter over buckets, feet	28.5	22.42
Polygon, diameter, feet	25.17	19.08
buckets, number	11	9
width, feet	4	3·75
length, feet	12	12
dip, feet	5.17	4·75

The following are the data of the run :—

	City of Erie.	*Tashmoo.*
Steam pressure, pounds	120.4	175*
Vacuum, inches	23.5	22
Air pressure, inches	5 max.	—
Revolutions per minute	33.25	40.08
M. E. P. in H. P. cylinder, pounds	58.4	—
I. P. cylinder, pounds	—	—
L. P. cylinder, pounds	36.6	—
referred to L. P. pounds	53.07	—
I. H. P., H. P. cylinder	2,004.6	—
L. P. cylinder	4,467.7	—
Total	6,472.3	3,400†
I. H. P. per square foot of grate	25.68	11.58
H. S. per I. H. P., square foot	1.79	2.57
I. H. P. per 100 square feet of wetted surface	50.66	39.1
Distance, in statute miles	94	94
Speed, in statute miles	21.76	21.70
in knots	18.899	18.847
Slip of wheel over bucket, per cent	35.65	32.2
Depth of water in feet, start	57	57
minimum	35	35
maximum	77	77
average	61.31	61.31
Admiralty coefficient	178.2	225

The trial resulted in the City of Erie beating the Tashmoo 45 seconds in a run of 94 miles—less than half a second a mile.

With the exception of four minutes' stop of one of the blowers on the City of Erie, the main engines and auxiliaries on both steamers worked perfectly throughout the trial; all journals running cool. The boilers furnished a full supply of steam at all times.

The admiralty coefficient should be considered in connection with the shallow water in which the run was made. It is estimated that the speed would have been at least one-quarter of a mile an hour greater had the steamers been in water of sufficient depth not to be influenced by the bottom.

I regret that owing to leaking snifting valves no indicator cards could be taken on the Tashmoo; the power was estimated from a previous trial, when 3,120 horse-power was developed, with thirty-nine revolutions. Plate No. 6 shows the cards taken on the City of Erie. Plate No. 5 graphically illustrates the performance of the City of Erie, showing the speed, horse-

* L. P. pass-over valve half open during last one and a half hours.

† Estimated.

power, slip, and depth of water at every moment and mile along the whole course.

Both steamers were taken off the regular service, and prepared for the trial in a few hours. The City of Erie arrived in Cleveland from her regular run from Buffalo only four hours before the trial commenced. She was not dry-docked to clean bottom, though that was in fair condition. Her boats were taken in on main deck to reduce wind resistance. The Tashmoo was dry-docked, bottom cleaned and painted. Both steamers were run by their regular crews.

DISCUSSION.

THE CHAIRMAN :—Before proceeding to the formal discussion, I beg the Society's indulgence to supplement Mr. Kirby's paper by a few remarks of my own, because, of course, I was naturally very much interested in the City of Erie's performance.

Mr. Kirby, in his paper, has given you only the solid facts of the City of Erie–Tashmoo race, without embellishment, and without calling your attention to many of the features of an interesting occasion.

The race was the unusual result of steamboat talk as to who possessed the fastest steamer on the Lakes. It was an unusual result in that a race occurred at all, and more unusual because all the care of an international yacht race was taken to make the affair satisfactory and decisive. Impartial judges and timekeepers were selected. Government inspectors were present, that no infraction of the law occurred. Two lanes were mapped out on the course, so that the racers were always at least a half a mile apart broadside, from the start to the finish, the lanes being marked at the beginning, the middle and the end of the course by tug-boats, duly anchored in place. The choice of position was decided by lot, the Tashmoo winning and wisely choosing the outer course.

Mr. Kirby has taken as a basis for the calculations in his paper, the distance of the course to be 94 statute miles. At the same time he correctly states that "computed from the latitude and longitude of the light-houses at Cleveland and Erie, and corrected for location of starting line, the distance is 94.31 miles." You will concede that no boat could sail the course in less than this distance. Again, he says, "Measured on the chart issued by the Hydrographic Office of the Navy Department, the distance is 95.25 miles." Other authorities make the distance 94¾ miles.

Those present at the race know that the boats each varied from the straight line, and I have for my calculations on the race, called the course 94½ statute miles. I trust you will agree that this is a fair assumption. You will note that the City of

Erie is a freight as well as a passenger boat, with a double tier of staterooms and appurtenances for night travel, designed to carry loads from 300 to 600 tons, and with wheels so placed that when the boat is loaded the power of the engine can be properly utilized, the immersion of the wheels being made to suit. At the time of this race, in order to have the wheels sufficiently immersed to utilize the power to a moderate degree even, it was necessary to carry a load of 250 gross tons.

Even with this handicap weight, considerable power was wasted, as is shown by the power per ton of displacement, by the high number of revolutions per minute, and the corresponding slip of the wheel. The slip of wheel is given for both boats taking the diameter given for outside of buckets. This is not correct, of course, because the path of the outer edge of buckets through the water is not a circle of the diameter given for the wheel, but this method of determining slip answers for a comparison. The Tashmoo slip is less than that of the City of Erie, showing a better hold on the water for the power developed. In this, however, the showing is not bad for either boat compared with fast side-wheel boats in Europe. The North Sea steamer, Marie Henriette, on trial, with 52.89 revolutions per minute, and 22.2 knots per hour, shows a slip of over 39 per cent. calculated by the method used by Mr. Kirby. If we could possibly have lowered the shafts of the City of Erie to give the wheels proper immersion without carrying that extra 250 tons load, the result would have been, of course, much more favorable for the City of Erie, but they had to take the "cart horse" just as she is against the "trotter," and make the best of it.

I now desire to call your attention particularly to the effect of the depth of water on the speed of the boats. When the race was first projected, the owners of the Tashmoo wished to have the race on Lake Huron where the water is considerably deeper than in Lake Erie, but the owners of the City of Erie preferred Lake Erie, and the choice was almost disastrous. You will remember that the Tashmoo chose the outside course where the water is from two to four fathoms deeper than on the inner course left to the City of Erie.

For the first hour and a quarter the boats ran neck-and-neck, any advantage for either being almost impossible to determine. At the expiration of this time I went to the pilot house to look at a Nicholson Patent Log, which shows the speed per hour being made by the boat, as well as making at the same time a chart of the speed. To my consternation, the pointer on the dial commenced to rapidly fall from 21.6 miles per hour, and as I had just left the engine-room a few minutes before, with engine running all right, making 33 revolutions, I at once jumped to the conclusion that something had happened to reduce the speed of engine, and I hastily returned to the engine-room to find the engine close shut off and making almost 34½ revolutions, indicating shoal water. A look out of the gangways proved this to be correct by the sea following on the quarters.

The Tashmoo, at this time in deeper water, immediately commenced to gain on the City of Erie rapidly, and held the gain for some time. As soon as the City of

Erie reached deeper water, however, she commenced to gain on the Tashmoo, but to those on the City of Erie it seemed a long, long time before the Erie regained what had been lost. It was a full half hour before the 21.6 mile mark on the dial was again touched by the hand, and not until 70 feet depth of water was reached did the City of Erie attain the highest speed of the race, 22.2 miles per hour. Mr. Kirby's diagram of curves of speed, power, slip, and of depth of water, shows at a glance the effect of the shoal water. This makes it very clear that had the race been in deeper water, the difference at the end would have been more than 45 seconds; but "all's well that ends well."

You will note that the City of Erie had not been dry-docked, and had arrived in Cleveland from Buffalo on her regular run with passengers and freight only four hours before the race commenced. So far as the engine and boilers are concerned, there was absolutely nothing done to prepare the boat specially for the race, because there was nothing needed to be done. The officers and crew had everything in proper condition, without any alterations or additions. The City of Erie went from regular business to a great trial, returned the 95 miles to Cleveland, and in a short time—as soon as passengers and freight could be placed on board—left for Buffalo again on regular business. Machinery designed for a maximum of 4,500 horse-power, developing under these conditions a maximum of 6,800 and an average of 6,400 horse-power, has, like Mark Twain's Jumping Frog, "good points." The changes in Mr. Kirby's table that I suggest, are as follows:—

	City of Erie.	*Tashmoo.*
Revolutions per minute	33.254	40.08
M. E. P. in H. P. cylinder, pounds	58.44	—
" L. P. cylinder, pounds	36.625	—
" referred to L. P. pounds	53.08	—
I. H. P., H. P. cylinder	1981	—
L. P. cylinder	4436	—
Total	6417	3400
I. H. P. per sq. ft. of grate	25.46	11.58
H. S. per I. H. P., sq. ft.	1.805	2.57
I. H. P. per 100 sq. ft. of wetted surface	50.23	39.1
Distance in statute miles	94½	94
Speed, in statute miles	21.88	21.70
" in knots	19	18.847
Slip of wheel over bucket, per cent	35.32	32.2

The difference in indicated horse-power is caused by my deducting the piston rods, which Mr. Kirby has not done. The diameter of H. P. rod is 7¾ in. and L. P. rod 8¾ in.

Let me call your attention to one more interesting item concerning the City of Erie, mentioned in Mr. Kirby's paper, which has nothing to do with the race.

Amount of coal used per round trip, 350 miles, 175 miles each day, 67½ tons of 2,000 pounds each. Amount used for same round trip made in 24 hours, 55 tons.

This latter record was made during the summer months of this year when these boats, City of Erie and City of Buffalo, made regular day and night trips between the cities of Cleveland and Buffalo, at an average speed of about 18½ miles per hour. There are no indicator cards showing the exact power developed, but 3,300 or 3,500 horse-power must have been necessary, which makes a small consumption of coal per horse-power per hour.

The difference between the 55 tons and the 67½ tons per round trip is caused by keeping up steam all day in port, when only a night trip is made, and the moving of boat from place to place at Cleveland to get freight and coal on board. Coal is now put on board without moving the boat from her own pier.

Mr. Kirby has sent photographs of the race, which are very interesting, and which I will leave here if you wish to look at them. He has also sent the original chart of this log which was in the pilot-house of the City of Erie, showing the speed at all times, and that is very interesting. If you care to look at that, you may do so. That is all I have to say on this subject, gentlemen. It is now open to you for discussion.

MR. ANDREW FLETCHER, JR., *Member:*—The illustrations accompanying the paper show, on board the City of Erie, a great number of people, and it occurs to me that the members might get the impression that the race was made under those conditions, while we who were there at the race know that that was not so; we were boiled down pretty well. Life-boats were taken off, and we even trailed the boat davits aft, and our colors were off; we flew about a fifteen-cent flag, and I might say that was about the same way with the Tashmoo; we were all locked in practically, even the reporters, and they had the hardest work. The windows on both boats were all closed. Both boats were stripped right down to their racing condition. I thought possibly it might be the impression here that we had a great number of people on board. While there were undoubtedly many people on board, the crowd was nothing like what these illustrations would show.

MR. WILLIAM H. FLETCHER, *Member:*—I think, Mr. Chairman, the thanks of the Society are due to Mr. Kirby for his paper, which, it seems to me, is an able presentation of the matter, and contains a great deal of valuable information. The remarks of my cousin just now in regard to being locked in might be supplemented by saying that we were not even allowed to go out of doors without a pass. But while we were well stripped on the City of Erie, we did not put any *weight* ashore. They had to put weight on board. I think one of the most striking things to me in the paper was the falling off in speed of the City of Erie, at the time you mentioned,

Mr. Chairman, when we were off Fairport. I was up in the Captain's room with you at the time, if you remember. My principal occupation was watching this speed curve. The City of Erie suddenly dropped off a mile an hour on that indicator, and when you went to the engine-room I went outside to the rail and saw this wave, usual in shallow water, following the boat. When we consider that at the time this speed drop of a mile an hour occurred, there was, according to the chart, about six fathoms under this vessel, drawing a little more than ten feet of water, it is rather remarkable that she should feel the bottom so decidedly with so much water under her. The other boat, according to the chart and according to the distance from our boat, had nearly two fathoms more water under her at this time than we did. I thought just before we reached Fairport, as nearly as I could judge from watching the position of the two boats, that we were as far ahead of her at that time as we were at the finish; although we crossed the line slightly in advance, that was due to our gathering way quicker than the Tashmoo did, as we started the City of Erie from the drop of the flag. It was a magnificent race. No race could be conducted more fairly, and no greater precautions could be taken to make the race sure and absolute than were taken in this case. I do not believe any one ever saw a race equal to it; certainly no one ever saw a better one.

MR. W. IRVING BABCOCK, *Member of Council*:—The general impression on the Lakes at the time of that race was that while the City of Erie was stripped the Tashmoo was not. I am a little surprised to hear Mr. Fletcher say that both boats were properly stripped. I was not there; I only know what I heard up on the Lakes—that the Tashmoo was beaten because they did not take proper precautions. Another thing that I notice is that the air pressure on the Erie is given as five inches as the maximum, which is pretty high. Was there any change indicated by this, or were there any unusual means taken to get vacuums?

THE CHAIRMAN:—On the Erie?

MR. BABCOCK:—Yes.

MR. W. H. FLETCHER:—I would like to answer Mr. Babcock, in a measure, on that point. I would say this:—that Mr. Andrew Fletcher and I arrived in Cleveland about 7.30 in the morning. We went down on board the boat, the Erie, a little before nine o'clock—about half-past eight—as I remember it. She was then discharging some of her freight. There was absolutely no preparation made as far as the boat was concerned, except to take the life-boats from above and put them on the main deck; they were not put ashore. There was nothing done around or about the engine, as far as I could discern, and I believe all who were on board will bear me out in this. Of course, we had good coal; that was evident. There is no question about the vacuum. I think if Mr. Babcock will read between the lines he will see some reason why cards could not be taken. Another thing that I noticed—I only

bring this out for discussion now as long as you mentioned it—was that it appeared to me from the Erie that the Tashmoo would come up on us and drop back, come up and drop back. Of course, I was not on the Tashmoo. I had never seen her except at a distance, but I had the impression at that time that they were using the pass-over up to the point when she would begin to lose vacuum, and then shut it off, and she would drop back. I may be mistaken. That is only a theory.

MR. F. P. PALEN, *Member*:—This trial between the Tashmoo and the Erie has developed a little point which is, I think, of very decided interest to all who are engaged in running trials, and that is this matter of depth of water and the influence on speed. I rather wish we had a little more information on it than this. I was particularly impressed by the remark of one of the gentlemen a moment ago, who said that he was surprised that six fathoms of water under the ship would have an influence. I remember an occasion when making a voyage on a ship from New York to Savannah, when we were taking very accurate data, and the depth of water ran all the way from twelve to sixty fathoms, by indicating the engine and keeping track of the revolutions and the speed of the ship, we could fairly accurately plot the depth of the water. This was a ship of about 6,000 tons displacement.

A MEMBER:—Was she a screw?

MR. PALEN:—Yes.

A MEMBER:—What was the speed, please?

MR. PALEN:—About seventeen knots.

MR. F. L. DUBOSQUE, *Member*:—There is very interesting data here in reference to the Howden system, giving dimensions of the blower, and so forth, and as this seems to have been a very good performance for the Howden system, I would suggest that the speed of the blower be given to complete the data.

THE CHAIRMAN:—Are there any other remarks? If not, with your permission, I will touch on some of the points that have been raised by the various speakers. As to the matter of these illustrations, they were put in the paper at my suggestion simply to show the character of the boat. I think the criticism of Mr. Andrew Fletcher, Jr., is a good one. You are not to understand that we were in that condition on the day of the race. The City of Erie was stripped to the extent of taking the life-boats down from the upper deck and placing them on the main deck. The flag-poles were taken down, and that is the only stripping that was done. That, however, was, I think, quite an important item, and that was done at the instance of the captain of the boat himself. The Tashmoo did not have her life-boats taken down nor her flag-poles. That is the only lack of strip that she had. She came

there prepared for the race, and, as I have been credibly informed, the engine was specially prepared for this race to the extent of trying to make her condensing apparatus more efficient to take care of the increased pressure on that engine. In other words, it was very much easier for those on the City of Erie to get a very great increase over the normal horse-power than it was on the Tashmoo. I perhaps would not have mentioned this fact had not Mr. Babcock asked if there was any special work done on the condenser of the City of Erie. There certainly was not. She took care of the steam perfectly well, and, as you see by the vacuum mentioned, sufficiently well for the requirements of the day in spite of the great increase in the horse-power. On the Tashmoo, however, I understand they covered the condenser with a special coating, and then played water upon that from the steam-pump in order to increase the efficiency of the condenser. Well, of course, that naturally made a mussy engine-room and made considerable vapor in addition to the leaking of the snifting valves, and that was why they did not take cards. On the City of Erie everything was in such perfect order that anybody could go to any part of her engine or boiler-room at any period of this trial without trouble. The engineer, and most of the time myself, stood in front of the engine taking observations. I kept a careful log of the performance of the engine every five minutes, and we had positively nothing unusual to do. There was no trouble in taking the cards any more than there would be on her regular trips. But that is all due, of course, to the style of engine, which is very well adapted for that sort of work.

As to the dropping back that Mr. William H. Fletcher refers to, I think that is due to variations in the course of the boats, rather than to any variation in speed. I am informed, and I believe that the information is correct, that the pass-over valves on the Tashmoo were not opened until the last half hour. Mention of that is made in the paper. But when two boats are traveling side by side, if one varies the course in one direction and the other in an opposite direction, the result would be an apparent going forward and dropping back. But, as I said before, during the first part of the race, it was very difficult to decide which boat had the advantage.

MR. W. H. FLETCHER:—It has been suggested to me that the condensers on both boats were jet condensers.

THE CHAIRMAN:—As to the speed of the blowers on the City of Erie, I think that is impossible to tell. That was one of the minor points we did not pay much attention to. We did all we could. That is all there is about that. The effort was to get there, and we did. No notice was taken of small matters like blowers. Most of the blowing was done afterwards. Speaking of that reminds me of an interesting incident that occurred as we were going over the Cuyahoga river that morning. Cleveland being the headquarters of the City of Erie, she was saluted in great style by every tug-boat and every boat in the harbor all the way out, and there was a great fuss made by every boat there. One of the firemen who stood near me said,

"That is all very well, but I hope they will be doing the same when we come back." It is needless to say they were.

As to the dip of the wheel on the City of Erie, Mr. Kirby and I talked that matter over last winter. We knew that it would be a very close race. Mr. Kirby was the father of both boats, and was sought by the managers of each of them to go with him on the day of the trip. But he wisely concluded that his best place would be on shore, because he was responsible for both boats, and he thought he would show no preference. After full consideration, Mr. Kirby and I came to the conclusion that about five and a half feet dip would be about the proper mean for the City of Erie. That is to say, we were working her so we would get about five and a half feet dip. I went up to Cleveland a day or two before the race, and made two trips in her between Cleveland and Buffalo, just to look her over, and I found there was a sentiment on board that we ought to run the boat as light as possible; so that, on the morning of the race, the Captain said to me: "How will you have this boat loaded? I will leave it to you." I said I would like to have five and a half feet dip of the wheels. Well, he thought that would make a great deal of load; "but," he said, "I will put it wherever you say." I had learned in the meantime that every man on board the City of Erie had bet on this race; I then made up my mind if I said five and a half feet dip and they put that load on and lost the race, I would have to walk home. So I compromised on the matter, and we put on a slightly less load than originally intended.

MR. W. IRVING BABCOCK:—Why did you reduce the load if a greater dip of paddle-wheel was more advantageous?

THE CHAIRMAN:—My life is still sweet, Mr. Babcock, and I did not know if I would even live if I lost that race by putting on load. We could have done a little better if we had put on more dip; that is, a little less waste of power would have resulted. But all is well that ends well. It was a beautiful race, and a great credit to both boats and the crews on the boats, for the responsibility was with them, after all.

If there are no further remarks on this paper, we will proceed to the reading of the second paper on the programme, by Professor H. C. Sadler.

MR. J. C. KAFER, *Member of Council*:—I second the motion of Mr. Fletcher that the thanks of the meeting be given to the author for his most interesting paper.

THE CHAIRMAN:—I certainly think that is due to Mr. Kirby.

The motion was unanimously carried.

Professor Sadler, the author, not being present, his paper on "Effect of Variation of Dimensions on the Stresses in a Ship's Structure," was read, in abstract, by the Secretary.

THE EFFECT OF VARIATION OF DIMENSIONS UPON THE STRESSES IN A SHIP'S STRUCTURE.

By Professor Herbert C. Sadler, Member.

[Read at the ninth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 14 and 15, 1901.]

The general increase in size of vessels, which has taken place within the past few years, has led to many interesting problems, among which those connected with the strength are perhaps the most important. The forces which act upon a vessel may be divided into two parts:—(1) Those which tend to strain it as a whole. (2) Those which tend to strain it locally. Of the first class those which cause bending in a longitudinal plane are the most severe, owing to the dimensions in this direction being the greatest. The general and local stresses are to a certain extent interdependent, the latter being often brought into existence by the former.

In all calculations respecting the strength of a vessel certain assumptions must be made as to the worst probable condition in which the vessel is likely to be placed, and also as to the way in which the accompanying bending moments will affect the structure as a whole. Hence at the outset it is well to remember that the calculated stresses are not necessarily absolute, but provided the conditions in each case remain the same these form a good basis for comparative purposes.

It is customary to calculate the stress due to bending from the formula $p = \frac{yM}{I}$, where p is the stress at a distance y from the neutral axis of the sectional area of the material at the point where M, the bending moment has its maximum value, and I is the moment of inertia of the sectional area of the material.

In a similar manner the stress due to a shearing force F may be calculated from the formula $q = \frac{FA\bar{y}}{It}$, where q is the shearing stress at any point, $A\bar{y}$ is the moment of the sectional area of material beyond that point, and t is the thickness of material.

The effect of variation of dimensions upon these stresses may be determined in a general way if certain assumptions are made. The bending moment will, in general, vary as the product of the displacement and length, and this is only true if the weights of hull, machinery, coal, and cargo vary as the cube of the dimensions and the distribution remains similar. The weights of machinery and coal do not vary in this ratio, and in most cases will vary more nearly as the square of the dimensions for a given type of vessel; but as these weights, in the case of a fast passenger steamer, are not more than from 35 per cent. to 40 per cent., and, in vessels of the intermediate type, from 15 per cent. to 20 per cent., and, in cargo vessels, from 5 per cent. to 10 per cent. of the total displacement, the error involved in the above assumptions will not be very great, provided that the variation in size from the type ship is moderate in amount.

In discussing the question of variation of dimensions upon the stresses in a ship's structure, it is also necessary to take into account the variation in the ratio of these to one another, if any such variation occur. The problem then divides itself into two parts:—(1) When all dimensions vary. (2) When one or two dimensions vary.

The following symbols will be made use of in what follows:—

L = length of vessel on L. W. L.
B = moulded breadth.
D = moulded depth to "strength" deck.
d = load draught.
t = thickness of material.
r = ratio of increase of dimensions.

If all dimensions vary, the displacement will vary as r^3 and the bending moment as r^4, provided the conditions of support remain similar; I will vary as r^4 and y as r. The stress will, therefore, vary as $\frac{r^4 r}{r^4}$, or as r, that is, the tensile and compressive stresses will increase as the dimensions increase. In the same way the shearing stresses will vary directly as the dimensions, for F will vary as the displacement, that is, as r^3, $A\bar{y}$ as r^3, I as r^4, and t as r; that is, q varies as $\frac{r^3 r^3}{r^4 r} = r$. The additional stresses due to pitching and heaving will vary in the same manner as the above, that is as the dimensions, provided the assumptions as to conditions are similar.

If the draught remain constant, which is practically what occurs at the

present day with all vessels over, say, 480 feet in length, and all other dimensions vary as before, the displacement will vary as r^2 and the bending moment as r^3, I and y will vary as in the previous case, and the stresses p and q will, therefore, remain constant. It will be seen, however, that the weight of hull increases as r^3, while the displacement increases as r^2, hence, in most cases, the bending moment would, in all probability, be relatively reduced as size increases, owing to the relatively smaller amount of cargo.

The effect of variation in ratio of dimensions amongst themselves upon the tensile and compressive and shearing stresses is shown in the following table for constant and varying draughts. It is assumed that when a horizontal or vertical measurement varies that the thickness of material in that direction varies in the same ratio :—

TABLE I.

Dimensions which vary.	Variation of Stresses.		Variation in Thickness of Material.	
	At varying draught.	At constant draught.	Horizontal.	Vertical.
L, B & D.....	T & C r Sh r	constant constant	r	r
L & B.....	T & C r^3 Sh r^3	r^3 r	constant	r
L & D.....	T & C r Sh r	constant constant	r	constant
B & D.....	T & C $1/r$ Sh constant	$1/r^3$ $1/r$	r	r
L	T & C r^3 Sh r^3	r^3 r	constant	constant
B	T & C r Sh r	constant constant	constant	r
D	T & C $1/r$ Sh constant	$1/r^3$ $1/r$	r	constant

T & C = tensile and compressive stresses; Sh = shearing stresses.

The relative importance of the various dimensions from a strength point of view can be immediately seen. It is interesting to note that under the given assumptions, with varying draught, provided the ratio L/D remains constant, the stress will vary as the dimensions, or as r, whether the ratio B/D is constant, increases, or decreases; if the ratio L/D increases the

tensile stress will vary as r^3 and the shearing as r^2, whether the ratio B/D is constant or increases; and if the ratio L/D decreases the tensile stress varies inversely as the dimensions and the shearing is constant whether the ratio B/D is constant or decreases. At constant draught if the ratio of L/D remains constant the stresses remain constant, whether the ratio B/D is constant, increases, or decreases; if the ratio L/D increases the stress varies as the square of the dimensions whether the ratio B/D is constant or decreases; and if the ratio L/D decreases, the tensile stress varies inversely as the square of the dimensions and the shearing inversely as the dimensions, whether the ratio B/D is constant or decreases.

In a paper read before the Institute of Naval Architects in England, Mr. John has shown that in vessels from 100 to 3,000 tons displacement the stress increases as the tonnage increases, and is, in fact, approximately proportional to the cube root of the tonnage, or, in other words, proportional to the dimensions. The question then arises, what is the limit of stress which should be allowed under the given assumptions as to support, and if necessary should the scantlings vary at a greater rate than the first power of the dimensions.

Following the same reasoning as above, it will be seen that for constant stress the moment of inertia should vary as r^5, that is, that the thickness of material should vary as the square of the dimensions. This involves a modification in the laws of similarity of bending moments in that the weight of hull would vary as the fourth power of the dimensions, and a point would soon be reached where the vessel would only just carry her own weight. The bending moment would generally decrease owing to the increase in weight of hull. It rarely happens, however, that one vessel is an exact reproduction of another but on a larger scale. With increase of dimensions, stability, as measured by metacentric height (BM), increases directly as the dimensions if all dimensions vary, and as the square of the dimensions if the draught remain constant. Hence as size increases the ratio of B/D does not necessarily remain constant, and for good sea-going qualities should decrease; and as shown above, any variation of this ratio has no effect upon the stresses provided the ratio L/D remains constant.

For constant stress, provided the variation in dimensions is not too great, the variation in thickness of material may be seen from the following table:—

CONSTANT STRESS.

Dimensions which vary.	VARYING DRAUGHT.		CONSTANT DRAUGHT.	
	Horizontal plates.	Vertical plates.	Horizontal plates.	Vertical plates.
L, B & D.....	r^3	r^3	r	r
L & B.....	r^3	r^3	r^3	r^3
L & D.....	r^3	r	r	constant
B & D..	constant	constant	$1/r$	$1/r$
L..	r^3	r^3	r^3	r^3
B	constant	r	constant	r
D..	constant	$1/r$	$1/r$	$1/r^3$

From the above table it will be noticed that, for constant draught, the thickness of material need not vary at a greater power than the first power of the dimensions, provided the depth varies as the dimensions; and in cases where the length does not vary, thickness of the material may be reduced. The part of the table for constant draught illustrates the case of large vessels of the present day.

The whole of the foregoing deductions have been based upon the assumption that as the size of vessel increases, the dimensions of the wave upon which she is supposed to be instantaneously poised, increase in the same ratio. So far as general theorizing goes, this is substantially correct up to a certain point. When a point is reached, however, where it is certain that waves of the assumed dimensions do not exist, the general theory no longer holds. That is to say, when we have reached our limiting ratio of wave height to length, as size is increased beyond this point the conditions of support will become gradually less severe, and hence the stresses upon the structure will be relatively reduced.

The assumptions usually made as to the worst condition in which a vessel of the ordinary mercantile type may be placed, are that she is instantaneously poised upon the crest of a wave whose length is equal to that of the vessel, and whose height is $^1/_{20}$ of that length. Observations of Lieutenant Paris and others have established the fact that as the length of wave increases the ratio of height to length decreases. In waves of 300 feet to 350 feet this ratio is approximately 1 to 20, and up to waves of 500 feet

it seldom if ever exceeds 1 to 25, while for waves still longer, these figures may drop as low as 1 to 50. Waves greater than 30 feet in height are not commonly encountered, and this height in conjunction with a length of 600 feet, corresponding to the ordinary assumptions as to wave height, may be considered as practically never occurring. Hence it seems scarcely reasonable to assume that a vessel of from 600 feet to 700 feet long would ever be instantaneously poised on the crest of a wave of its own length, and from 30 feet to 35 feet in height.

In the same way with regard to pitching, the probability of a vessel of this length falling in with a series of waves of the above dimensions is exceedingly remote, and decreases rapidly as the length of ship increases. Hence it is reasonable to suppose that the bending moment upon a vessel will not increase as the fourth power of the dimensions when the larger size is reached, and, as will be shown later, will vary more nearly as the cube of the dimensions.

In dealing with bending moments for comparative purposes, it is convenient to refer them to the product of the displacement and length, the factor taking the form, $F = \frac{\Delta x L}{M}$ (where M = bending moment and Δ = displacement).

On Plate 7 the results of a number of calculations upon actual vessels of varying types have been plotted, and it is interesting to note that, although these vary from passenger to cargo vessels, the value of the factor F, as represented by the cotangents of the angles BAD and CAD, lies within comparatively small limits, the fine vessels in general lying near the upper line and the full ones near the lower. In the above calculations each vessel was assumed to be instantaneously poised upon the crest of a wave of its own length and one-twentieth of that length in height, and no account has been taken of the variation of pressures of different depths in the wave formation from that due to the corresponding depths in still water. The factor so obtained for any given type could be used with a fair degree of accuracy for different sizes, provided the variation was not of great amount, but for the reasons above stated this factor would in general be too low for the largest vessels. On Plate 9 the variation of this factor in terms of wave heights, as represented by percentage of the length, has been plotted for a few typical merchant vessels.

The value naturally falls considerably from the still-water condition to that where the vessel is poised on the crest of a wave whose height is five per cent. or one-twentieth of its length. From these curves the variation

in bending moment, and hence stress due to various assumptions as to wave heights, may be determined, the curves as drawn graphically representing the reciprocal of the stress, the sectional area and arrangement of material remaining constant.

For constant stress it was seen that, if the thickness of material varied as the dimensions, the bending moment would have to vary as the cube of the dimensions—that is, that the value of the factor F instead of remaining constant would have to increase with increase of dimensions. The question then arises, is it fair to assume that this is the case, and what does the assumption involve? Turning to Plate 9, let us take as an example the vessel whose factor F is represented by the curve AB. (The curves from which this has been obtained are shown on Plate 8.) This vessel is 530 feet in length and has a value of F approximately equal to 28 when on the crest of a wave whose height is 5 per cent. of its length. Suppose we have a vessel similar to the above, but 700 feet in length, and let us assume that the stress on the 530-foot ship is the highest permissible. In order that the stress may remain constant the value of F should be increased in the ratio of 700 to 530, or the new value for F will be, say, 37.0. From the curves this value corresponds to a wave height of, say, 2.9 per cent. of the length, or 20.3 feet. So that for constant stress, if the thickness of material vary as the dimensions, the conditions of support must be that the vessel is on the crest of a wave 700 feet long and 20.3 feet high. In all probability, this ratio of height to length is a fair one for a wave of this length. It should, however, be remembered that in passing from the 530-foot to the 700-foot ship the draught is assumed to vary as the dimensions. In the case of the 530-foot ship with the coal out, the draught was about 26 feet, so that the corresponding draught of the 700-foot ship would be 34.3 feet, and the fully laden draught about 37 feet. With the present limits of draught these are impossible, so that the bending moment, as obtained above, is in excess of that which would occur, at any rate, at the present day. In other words, if we allow the bending moment so obtained to be correct and the draught to remain constant, it is equivalent to the assumption that the wave height is increased. In the case cited the corresponding height would be approximately 25 feet. The stress would therefore remain constant, if the thickness of material varied as the dimensions only, the bending moment not increasing in any cases at a higher power than the cube of the dimensions. The weight of hull in this case will vary as the cube of the dimensions, and the curves in Figure 2 will hold for all sizes with a fair degree of accuracy. It may be of interest to see what size a vessel of this type would have to be

in order that the wave height should be reduced to zero. From the same curve as before, the value of F at zero height of wave is 71.5. The dimensions of the vessel would, therefore, have to be $\frac{530\ 71.5}{28}$ = say, 1,350 feet. So, that carrying the above reasoning out to the extreme limit, the condition of support causing the most severe bending moment on a vessel of 1,350 feet in length and say, 66 feet draught, would have to be assumed as the still-water condition. As, however, the draught considerations for some years to come will scarcely reach this amount, and any reduction in the above draught is virtually equivalent to an increase in wave height, the bending moment, as obtained above, might not be very far from the truth. The above is, of course, beyond the limits of present day practice, and is merely added as an illustration of the limits to the above reasoning. In general, as the larger sizes are reached a different arrangement of material becomes necessary. The height between decks does not necessarily increase as size increases, and hence the moment of inertia will relatively increase in the larger vessels. It was assumed in the case of the 530-foot ship that the stress was the highest permissible, and this stress, at the top of the shear-strake, amounted to 10.5 tons. There are many vessels afloat at the present day where this stress is equalled, if not exceeded, and, so far as the general structure is concerned, are perfectly satisfactory. This, or any other stress, will be maintained as size increases and thickness of material increases in the same ratio, if, as already discussed, the assumed conditions as to support are modified in the larger vessels to conform with the probable conditions in actual practice. Or, looked at in another way, if the conditions as to support are assumed to be exactly similar in all cases, *i. e.*, the dimensions of the wave vary as those of the vessel, the stress could be allowed to vary with the dimensions and the same degree of safety be obtained. That is to say, that Mr. John's table could be continued to the larger vessels indefinitely, because at the point where the maximum calculated stress occurs, say in vessels from 400 feet to 500 feet in length, the ratio of wave height to length reaches its maximum, and beyond that point this ratio steadily diminishes.

With the present style of construction the neutral axis of the section is usually below the mid depth, owing to the accumulation of material in the double bottom. Hence the value of y is, in general, about 60 per cent. of the depth, so that anything that can be done to decrease this will decrease the stress. This has been realized in some of the later large vessels by taking material from the lower decks and placing it in the upper or strength deck,

but the system might be carried to an even greater extent with advantage. The question as to which deck should be made the "strength" deck involves certain difficulties. Where a vessel is fitted with a long bridge house a great increase in moment of inertia may be obtained by making the bridge deck the strong one and the bridge deck shearstrake the main shearstrake. But the ends of such structures in large vessels are always giving trouble, for at the points of discontinuity there is generally an accumulation of stress which in some cases causes rupture of the plating, even though these points have been specially strengthened. However, with careful attention to details, this objection may be overcome and a satisfactory junction made.

Similar difficulties occur with promenade decks where these are fitted for practically the whole length of the vessel. Expansion joints help to relieve the stress on those decks by allowing them to strain, but the difficulty is to know where to stop the expansion joint. If the deck only be allowed to strain, this will throw a severe stress upon the deckhouse plating connected to it, and if this be cut in the same way as the deck above, the trouble is passed on to the deck below. The best method of construction would be to so arrange deckhouses and erections that they are independent of the main structure, but the practical difficulties prohibit the adoption of such a system. The only other way out of the difficulty is to cut the deckhouses wherever possible and fit them with well-rounded corners. Where discontinuity of structure occurs it is necessary not only to provide ample longitudinal strength, but also sufficient transverse strength to keep the longitudinal members to their work and prevent buckling. Attention should also be paid to the arrangement of butts so that none of these come too near any opening or discontinuity, and also efficient compensation provided in way of openings, so that the sectional area of material is at least equal to that where no openings occur.

The connection of the double bottom with the main frame of the vessel (another point of discontinuity) has required extra stiffening in the larger vessels. The style of double bottom as usually adopted, does not lend itself particularly well to an efficient arrangement, and from structural considerations some form such as that used in war vessels seems preferable as being of a more continuous nature.

Several of the larger steamers of the present day have shown signs of local stresses at the forward end, the bottom in some cases becoming corrugated between the frames, and in others being bodily set up. The cause is rather difficult to detect, but is no doubt due to pounding in a heavy sea. These parts can, however, be easily stiffened by doubling the frames and plating,

and as an additional security, increasing the riveting both in the landings and butts, for it is in the neighborhood of this part that the shearing stresses have the maximum value. In fact, the riveting in large steamers throughout requires very careful attention, more perhaps than is usually given.

Of late years large vessels have rarely shown signs of general weakness of structure, the difficulties experienced in some being due to local causes such as the reduction of material in the way of openings or at discontinuities. Because a vessel shows signs of weakness at one particular place, it does not necessarily follow that the structure as a whole is weak, and that, therefore, the thickness of material throughout should be increased. Local stresses should be met by local, and not general, strengthening.

The subject of the strength of ships is one which is continually before the shipbuilder, and this paper has been prepared mainly with the object of starting a discussion upon this important matter.

APPENDIX.

In performing calculations of bending moments one of the most laborious parts is the determination of the "weight-of-hull" curve. The method that has been adopted throughout has been to assume the total weight of hull distributed as shown in Figure 4, Plate 9. This no doubt involves a certain error, but when it is remembered that the weight of hull is only part of the total weight of the ship, and that the bending moment is obtained by the second integration of the load curve, it will be seen that the error is inappreciable, and for comparative purposes negligible. The following factors will be found to give fairly accurate results for all vessels of normal types. If H = weight of hull and L = length of vessel H/L = mean weight per foot of length :—

	Fine vessels.	*Full vessels.*
Bow ordinate	= .566 H/L	.596 H/L
Midship ordinate	= 1.194 H/L	1.174 H/L
Stem ordinate	= .653 H/L	.706 H/L

It is also convenient to bring all these calculations to one set of scales, so that the final result is independent of dimensions. The following will be found convenient, and, although perhaps known to most members, is reproduced here with the suggestion that if at any time any investigations of this character are carried out, the same scales be adopted, and hence something approaching a standard be maintained. Let the length of the vessel, L = 20 inches. Therefore, in all cases, the scale for length will be 1 inch = $L/20$ feet. Mean ordinate of weight or buoyancy curves = 3 inches. Therefore scale for weight or buoyancy:—1 inch = $\frac{\text{Dispt.}}{3\,xL}$ tons per foot. The total area of the weight or buoyancy curve will be 60 square inches. Let 1 inch ordinate of shearing force curve = 2 square inches of load curve. Therefore scale for shearing force:—1 inch = $\frac{\text{Dispt.}}{30}$ tons. Let 1 inch ordinate of bending moment curve = 3 square inches of shearing force curve. Therefore scale for bending moment:—1 inch = $\frac{\text{Dispt.}\ xL}{200}$. As the area of the buoyancy curve in all cases is 60 square inches, it will readily be seen that for preliminary investigations a few standard curves for different types will enable the above calculations to be performed without going through the somewhat laborious work of obtaining the buoyancy curve for each special ship.

DISCUSSION.

THE CHAIRMAN :—Gentlemen, the paper is now before you for discussion. Are there any remarks to be made ?

THE SECRETARY :—I suggest, Mr. Chairman, that we follow the usual procedure in such matters. There are possibly many who would like to add to the discussion on this and other papers who are unable to be present at the meeting, and I suggest that the usual privilege be extended of letting them send their communications to the Secretary to be embodied in the proceedings. Of course, a paper of this kind presents data which cannot be discussed off-hand very well, and, as we were very much delayed in the printing, it was not possible to circulate the papers much in advance of the meeting. I am sure that the Society is very much indebted to Professor Sadler for his interesting paper, and I move that a vote of thanks be extended to him, and that the privilege of submitting written communications be granted to those who may desire to avail themselves of it.

The motion was seconded by Mr. Forbes and carried.

THE CHAIRMAN :—The thanks of the Society are given to Professor Sadler for his able paper.

The next paper on the programme is "Power Consumed in Propelling the Whitehead Torpedo at Various Speeds," which will be read by the author, Mr. Frank M. Leavitt.

Mr. Leavitt read his paper.

POWER CONSUMED IN PROPELLING THE WHITEHEAD TORPEDO AT VARIOUS SPEEDS.

By Frank M. Leavitt, Esq., Member.

[Read at the ninth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 14 and 15, 1901.]

The Whitehead torpedo is too well known in its general features and has been too frequently described in various publications to require any special introduction here. It has for many years held a prominent place among the weapons of the principal navies of the world, the last one to adopt it being the United States. Some ten years ago this Government placed its first order for torpedoes with the E. W. Bliss Company, and, since that time, the latter has furnished all which have been used in the service.

There are many interesting details connected with the manufacture of these weapons, among which I invite your attention to certain tests which have been made by the writer with the view of ascertaining the efficiency of the motive power, and the amount of power required to drive the torpedo at various speeds up to thirty-three knots per hour.

The results of these tests, *per se*, may not be considered a very valuable acquisition to scientific lore. But the apparatus used in making them is quite novel and capable of being efficiently applied in other directions as, in fact, has since been done.

The trials have also led to certain deductions relative to the frictional resistance of smooth surfaces passing through water, which I believe have a permanent value.

The data here presented are the net results of a large number of tests upon several hundred torpedoes, so that ample opportunity has been had to check and verify the figures.

The trials have consisted of 1st, a series of shop tests of the motive power as measured by an absorbing dynamometer; and, 2d, runs made by the torpedo over a measured range under service conditions.

In the latter case, as the operation of the torpedo is beyond observation or control, it would be difficult, not to say impracticable, to attempt any direct measurement of engine performance. These service runs are made

for the purpose of fulfilling certain contract requirements, and the only data derived from them bearing upon the problem in hand are, 1st, the speed maintained over the range; and, 2d, the amount of compressed air expended in driving the torpedo at this speed.

By the shop test, the amount of energy exerted at the propeller shaft is determined for each pound of air expended by the engine.

By combining the results of these two series of tests, the shaft horse-power required to drive the torpedo at the observed speeds may be estimated.

Before proceeding farther it may be well to describe, in brief, so much of the torpedo mechanism as has to do with the subject in hand.

Three general types have been furnished to our Navy, known as 3.55 Metre Mark I and II, 5 Metre Mark I, and 5 Metre Mark II.

Their outlines are shown in Figures 1, 2, and 3, Plate 10. These diagrams are drawn to the same scale, and present to the eye the essential difference in design. A photograph of the short torpedo is reproduced in Plate 13, and of the long Mark I in Plate 14. All three have a diameter at the middle body of 45 centimetres, or a trifle less than 18 inches.

As is generally known, the motive power is compressed air. It is stored in a strong steel flask, which forms the middle body of the torpedo, and occupies about one-half its displacement.

In the short torpedo, and in the long Mark I, the maximum pressure of air carried is 1,350 pounds. The flask of the long Mark II is heavier, and carries a pressure of 1,500 pounds. The weight of air carried at these pressures varies somewhat with the weather conditions, but will average about 48 pounds in the short torpedo, 65 pounds in the long Mark I, and 80 pounds in the long Mark II.

The main engine is of the three cylinder Brotherhood type, taking air on one side only of the pistons. The three connecting rods work upon a single crank pin, and one cam upon the crank shaft operates in succession the three engine valves, which latter are of the piston type. The valves are set to cut off at one-third the stroke of the pistons.

Air is supplied to the engine through a copper tube leading from the air storage flask. There is interposed in the line of this pipe a starting or throttle valve, and in addition thereto a pressure reducing valve, which latter is so constructed as to cause a practically constant pressure to be maintained at the engine. An adjustment is provided by means of which the working pressure on the engine may be increased or diminished at will. Having a fixed amount of energy stored in the flask it is quite apparent that the greater the distance it is desired that the torpedo shall run, the

lower must be the working pressure on the engine. It is possible, therefore, to drive the torpedo at a high speed over a short range or at a lower speed over a correspondingly longer distance.

To regulate at will the distance which the torpedo shall travel, a device is attached to the throttle valve, operated by a train of mechanism from the crank shaft which automatically closes the throttle when the predetermined distance has been traversed.

There are two propellers, placed tandem, as clearly shown by the photographs. The after one is attached rigidly to the main shaft and the forward one to a sleeve, concentric with the shaft, but made to revolve in the opposite direction by means of a system of mitre gears. The propellers must, therefore, of course, be right and left-hand. The object of this arrangement is to neutralize the tendency which a single screw would have to rotate the torpedo bodily on its longitudinal axis. As it is entirely submerged and has a metacentric height of only about three-tenths of an inch it has not sufficient stability to withstand such a force.

As already stated, the two factors to be determined by the shop test are the weight of air passing through the engine, and the corresponding foot pounds of energy delivered at the shaft. The most obvious, and also the most accurate, way to arrive at the first is merely to weigh the air flask before and after the test, the loss in weight being the weight of air used. This method has frequently been resorted to, but as it involves considerable time and trouble in disconnecting the apparatus another plan has generally been adopted. This consists in attaching a pressure gauge to the flask to enable the pressures therein before and after the test to be noted. Knowing the cubic capacity of the flask, the weight of air can be calculated. This method is open to the objection that at any given pressure the weight of contained air will vary with the temperature. In the process of compression considerable heat is, of course, generated, and notwithstanding that the air is passed through coolers on its way to the flask, it is liable to arrive there at a somewhat higher temperature than the surrounding atmosphere. Furthermore, the work done during the test in expelling the air from the flask leaves that remaining at the end at a considerably lower temperature than the normal. In order to eliminate these errors the whole apparatus is allowed to stand after charging until the temperature has fallen so near to that of the atmosphere that no farther drop in the pressure gauge is perceptible, and likewise, after the run, the pressure is not noted until the gauge stops rising. The loss of pressure after charging the flask is generally not more than a few pounds, but, after the run is completed, the gauge

frequently runs up as much as one hundred pounds or more. In computing the weight of air by means of gauge readings, due allowance is made for variations in weight due to the shop temperature. These precautions being taken, it is found by comparison with the other method of direct weighing that the results are quite accurate and to be relied upon. In order to simplify the calculation, tables have been prepared giving the weight of the contained air for the various pressures and shop temperatures.

The other factor to be arrived at, viz., the energy delivered by the engine at the propeller shaft, is determined by means of an absorbing dynamometer attached to the shaft in lieu of the propellers.

For this purpose none of the ordinary forms of Prony brake were considered suitable. The duration of the test being limited to one minute or less, it would be difficult to obtain accurate measurements with so unwieldy an instrument. Recourse was, therefore, had to a plan suggested by Prof. Webb, of Stevens Institute.

His idea was that a number of smooth metal discs revolving in water could be made to absorb the power of the engine by their surface friction. The rotative tendency would be transmitted to the water, and through it as a medium to the vessel containing it. If, then, the latter were mounted in suitable bearings and free to turn concentrically with the engine shaft, by resisting this tendency with a spring balance, the torque could be measured the same as with a Prony brake.

At a given speed of rotation the frictional resistance would be constant, so that, by properly proportioning the areas of the discs, an engine of any given horse-power could be controlled at the desired speed of revolution.

Prof. Webb kindly furnished me with a sketch of his idea, but he did not go so far as to suggest the proper number of the discs nor the size they should be made, and I could find no published data to aid me in determining these rather important details. The torpedo itself, however, furnished the necessary information. The formula given below was based on the assumption that the entire thrust of the propellers is absorbed in overcoming the frictional resistance of the torpedo passing through the water, and that this resistance varies as the square of the speed. Also, that the same holds true in regard to the revolving discs.

Let V = velocity of torpedo in feet per second.
A = area of wetted surface of torpedo in square feet.
V_1 = velocity of rotation of discs in feet per second measured at a distance from the center equal to .707 × radius.
A_1 = total area of discs in square feet.
R = revolution of discs per second.
r = radius of discs in feet.
N = number of discs.

Then, if our premises are correct, we shall have

$$AV^2 = A_1V_1^2 \qquad (1)$$

and as

$$A_1 = 2\pi r^2 N$$

and

$$V_1 = 1.414\,\pi rR$$

we may put the equation in the form

$$AV^2 = 123\,r^4R^2N. \qquad (2)$$

The area of wetted surface of the short torpedo is 50 square feet, and its mean velocity about 46 feet per second, so that $AV^2 = 105800$, which, substituted in equation (2), gives

$$N = \frac{860}{r^4R^2}. \qquad (3)$$

It was desired that the dynamometer should be so proportioned that the speed of the engine should be controlled at about 900 revolutions per minute, or $R = 15$. Introducing this value in equation (3)

$$N = \frac{3.82}{r^4}. \qquad (4)$$

The size of discs adopted was two feet diameter, or $r = 1$, so that $N = 3.82$. That is to say, four discs each two feet in diameter, revolving in water at 900 revolutions per minute, should absorb fully as much or more power than is required to drive the torpedo through the water at a speed of 46 feet per second.

It will be observed that the combined area of four such discs is just about one-half the area of wetted surface of the torpedo. I did not have sufficient faith in the above calculations to trust entirely to them, and so built the first dynamometer with six discs two feet diameter in place of four. Upon attempting to use it, however, the reasoning was vindicated, and I found it necessary to remove two of the discs.

Plate 15 shows the general appearance of the first instrument constructed. The shaft was placed upright, and the engine to be tested was secured to a bracket overhead, and its shaft coupled to the dynamometer shaft. The latter entered a cast-iron casing placed near the floor and supported by a step bearing. The four steel discs, which were nothing more than ordinary circular-saw blanks, without teeth, were keyed to the shaft within the casing. A system of links and levers connected the casing with a spring balance shown in the photograph at the left overhead. A dash pot for the purpose of absorbing undue vibrations or fluctuations of speed was attached to the lower end of the rod connecting with the spring balance, but this was found to be quite unnecessary as the action of the apparatus is remarkably smooth without it.

The upper and lower disc cleared the inner faces of the casing by about $\frac{1}{8}$ of an inch, and the space between the discs was about $\frac{1}{2}$ inch.

In order to prevent the water contained in the casing and lying between the discs from partaking freely of the rotary motion, by which it was thought the frictional resistance would be largely reduced, a stationary disc, having its outer edge rigidly attached to the casing and an opening through the center large enough to amply clear the shaft was inserted between each of the revolving discs.

If the dynamometer were to be used continuously for any considerable length of time it would be necessary to keep a stream of water running through it, as otherwise the water would quickly boil. For the short runs made with the torpedo engine this precaution is not necessary.

It will be observed that the torque, as measured by the spring balance, varies as the square of the revolutions, and that the horse-power absorbed will be a function of the two.

It is, therefore, possible to construct a table, giving in three columns, respectively, the tension on the spring balance, and the corresponding revolutions per minute and horse-power absorbed. For illustration, a table is given below condensed from the one used in connection with this dynamometer:—

Tension in lbs.	R. per Min.	H. P.
26	802	15.8
30	862	19.6
34	920	23.8
38	974	28.1
42	1024	32.6
46	1072	37.3
50	1118	42.3

The original scope of the trials did not extend beyond making a comparison between newly-built engines and those already tested in service runs and known to possess the required efficiency to obtain contract speed. By the aid of such a table as the one above these trials were reduced to a shop routine conducted by the workman in charge of assembling the engines. To make the test he mounted the engine in place and piped it up to an air flask to which a pressure gauge was attached. After noting the gauge pressure he would start the engine at full speed and allow it to run until the pressure in the flask had fallen to about one-half the original pressure, meantime noting the reading of the spring balance. The only mathematical calculation then required of him was to divide the number of pounds drop on the gauge by the noted tension on the spring balance. The result must at least equal a constant which represented the minimum efficiency allowable. If the result fell below this amount the engine was set aside for readjustment. In order to check the result and to know that the dynamometer was in proper working condition a tachometer was attached to its shaft, and the reading of this must correspond with that given in the table opposite the observed tension on the spring balance.

A general formula for this type of dynamometer has been deduced as follows:—

$$H.\ P. = \frac{TV_1}{550} \times \text{constant.}$$

From formula (2) above, the thrust $T = r^4R^2N \times$ constant, and as $V_1 = \pi rR \times$ constant we can write, after reducing,

$$H.\ P. = \frac{r^5R^3N}{C}. \qquad (5)$$

For general purposes it is desirable to have the shaft of the dynamometer horizontal in place of vertical, and several have been constructed on this

plan. By making the instrument of sufficient capacity it is not necessary that the casing should be entirely filled with water. As a matter of convenience it is best that the water level, when the shaft is at rest, should be below it, thus avoiding the necessity of stuffing-boxes. Under these conditions, as soon as the shaft attains any considerable speed the water is thrown outward and forms a ring entirely around the periphery of the discs. This state of affairs can be taken care of in the formula by writing

$$H.P. = \frac{(r^5 - r_1^5)\ R^3 N}{C} \qquad (6)$$

in which r_1 is the inner radius of the wetted surface.

The horizontal shaft arrangement has an incidental advantage over the vertical in the ease with which various overflow openings for the water may be provided in the casing at different distances from the bottom. By admitting more or less water the resistance at any given speed may be varied so as better to adapt the instrument to general use.

The value of the constant C may be placed at 600, as I have built several dynamometers in which this condition was fulfilled. It is evident, however, that it would vary to a greater or less degree with the nature of the frictional surfaces.

The following table is based on the formula with $C = 600$:—

HORSE-POWER ABSORBED BY A SMOOTH STEEL DISC REVOLVING IN WATER.

Diameter of discs.	Revolutions per Minute.						
	600	700	800	900	1000	1100	1200
Inches.							
24	1.67	2.64	3.95	5.62	7.71	10.26	13.33
30	5.09	8.06	12.04	17.16	23.52	31.32	40.69
36	12.65	20.06	29.98	42.71	58.52	77.93	100.15
42	27.36	43.37	64.8	92.33	126.5	168.4	218.9

I wish to call especial attention to the great influence which a small increase in the diameter has upon the resistance. It will be observed, for instance, that a disc three feet in diameter will require to drive it at 1,200 revolutions about 100 horse-power, while if its diameter be increased by

six inches the power is considerably more than double to maintain the same speed.

The diameter of the propellers of long Mark II torpedo is about 14 inches, and their area about $^1/_5$th that of a disc of the same diameter.

Applying the formula we find that at a speed of 1,000 revolutions per minute, the power absorbed in overcoming the friction of the two wheels is about one-fourth of one horse-power, an amount not worth considering. Suppose, however, that we were to design a torpedo driven by a turbine engine, where the engine speed would have to be about 6,000 revolutions per minute, and that we coupled the propellers direct to the engine without gearing. We find that a single disc 12 inches in diameter, or two discs 10½ inches diameter, would absorb 50 horse-power in overcoming surface friction, the whole probable power of our engine, and propellers of like diameter would waste something like 20 per cent. of our available power. At ordinary speeds, therefore, the surface friction of propeller blades may be neglected. At the high rate of rotation required for boats driven by a turbine, surface friction becomes an important factor, necessitating the use of screws of small diameter. The above data will enable the loss from this source to be calculated.

The second, and more elaborate dynamometer constructed is shown in Plate 16. The object in this case was to test the motive power and its regulating devices as a whole under conditions as nearly approaching those of actual service as could be had in the shop.

For this purpose, therefore, the torpedo was wholly assembled, with the exception of the propellers, and placed bodily in the tank shown in the background of the photograph, where it was entirely submerged in water. The rear end projected through the end wall of the tank so as to enable the propeller shaft to be coupled to the dynamometer. The latter was furnished with a single disc 34 inches in diameter.

The casing was supported on roller bearings, and the spring balance for measuring the torque was attached directly to it. In order to obtain a graphic record of the test, a drum was provided so geared to the dynamometer shaft that its surface advanced $^1/_{32}$ of an inch for each revolution of the shaft. A vertically moving pencil traversed the face of the drum, operated by the movement of the casing, so that the vertical height of the line above the zero line measured the torque at each instant of the test. For the purpose of recording time a second pencil was placed directly under the other. It was operated by an electro-magnet in circuit with a pendulum beating seconds, and made a gap on the card at the end of each second. A typical card taken

in this way and copied on a reduced scale from an actual run is shown in Figure 8.

The oscillations noticed at the beginning are due to the suddenness with which the engine is started, throwing the weight of the casing against the spring balance, the undulations dying out after a few seconds. The other irregularities in the line are due to the fact that the reducing valve does not maintain an absolutely uniform pressure on the engine throughout the run, and the dynamometer responds to very small changes in the power transmitted to it. In making this test the engine is started by quickly opening the throttle wide and thereafter leaving the apparatus entirely uncontrolled, save by its own mechanism, until it stops automatically.

The area of the card taken in the manner above outlined gives the total energy delivered to the dynamometer from beginning to end of run. Dividing this by the weight of air used in obtaining it gives us the energy expended for each pound of air used. The general average of some hundreds of tests has established the fact that these engines, run under the conditions given, will exert 25,000 foot pounds of energy for each pound of air expended. The performance never exceeds this amount by more than 4 per cent., and rarely by more than 1 or 2 per cent. When an engine is found to fall short of this amount by more than 1 or 2 per cent. it is sent back for readjustment.

At the beginning of a run, whether in the shop or in service, the temperature of the air in the flask is practically that of the surrounding atmosphere. In doing its work, however, its temperature is greatly reduced, the exhaust being extremely cold. If heat is imparted to the air during the run the available energy will, of course, be proportionately increased. Until recently no successful attempt has been made in this direction owing to difficulties of both a mechanical and military nature. During the past two or three years, however, I have done considerable experimenting along this line, and within the year have furnished the Navy with a torpedo equipped with a superheater. This device is arranged to start combustion of a liquid fuel within the air flask at the instant of launching the torpedo. The tests show a net gain of about 40 per cent. in the energy transmitted to the dynamometer, the work done being about 35,000 ft. lbs. for each pound of air expended as against 25,000 when the superheater is not in operation.

There is an additional incidental gain in the use of the superheater due to the fact that at the end of the run the air remaining within the flask being warm has less weight at the same final pressure than when cold, so that a greater percentage of the whole amount stored will have been used.

The net gain in speed of the torpedo due to the superheater is nearly five knots per hour.

A torpedo equipped with this device has been in the hands of the Navy Department for some months past, and although favorably commented upon has not, at the present writing, been adopted in the service.

In making the tests of the torpedo under service conditions over the measured range, the weight of air expended is computed in the same manner as in the case of the shop test already described. The speed is determined by noting, with the aid of a stop watch, the time elapsing from the instant of launch to the passage of the torpedo by the target.

Calling W the total weight of air used up to the target, the gross energy expended in driving the torpedo over the range will be, under normal conditions,

$$\text{ft. lbs.} = 25000\ W,$$

and where the superheater is in action

$$\text{ft. lbs.} = 35000\ W.$$

Therefore, if the duration of the run in seconds be denoted by t, we shall have, for the horse-power expended at the shaft,

$$H.\ P. = \frac{25000\ W}{550\ t} \text{ for cold air}$$

$$H.\ P. = \frac{35000\ W}{550\ t} \text{ for warmed air.}$$

The following table gives a summary of the results deduced from a large number of trials :—

Type of Torpedo	Range in yds.	Pressure carried in air flask.	Weight of air used during run.	Total work done by engine at shaft, ft. lbs.	Duration of run. Seconds.	Observed speed kts. per hr.	Shaft horse-power.	
			lbs.					
3.55 M. Mark I and II.	800	1350	30	750,000	53.3	26.66	25.6	air cold
5 M. Mark I.	800	1350	42.3	1,057,000	51.6	27.52	37.2	" "
5 M. Mark II..	800	1500	44.5	1,112,500	49.8	28.54	40.6	" "
" "	800	1500	51.6	1,806,000	42.4	33.5	77	" hot
" "	1500	1500	62	1,550,000	115.5	23.5	24.4	" cold
" "	1500	1500	63.4	2,219,000	96.3	27.6	41.9	" hot

The speeds and corresponding horse-powers given above have been plotted in curve shown on Plate 12 in order to graphically show their general relation. The curve drawn in the diagram is based on the formula

$$H.\ P. = CV^{3.5}.$$

DISCUSSION.

THE CHAIRMAN: Gentlemen, the paper is before you for discussion. We would like to hear from any member. If there is no one who cares to discuss this paper at the present time, the same privilege is open to members as in the case of the preceding paper—that is, to send to the Secretary any discussion which members may desire to submit on this subject. In the meantime, I will take it for granted that you desire to extend a vote of thanks to the author for his very valuable paper.

The vote of thanks was unanimously carried.

THE CHAIRMAN:—You will see that the next papers on the programme are the prize competition papers on the "Balancing of Marine Engines." The Chair has just been handed a memorandum, giving the names of the authors, and has the pleasure of announcing that the first prize paper, bearing the motto "Labor omnia

vincit," was presented by Naval Constructor D. W. Taylor, United States Navy (Applause). The paper bearing the title "Let Truth be our Guide," to which was awarded the second prize, was presented by Rear-Admiral George W. Melville, United States Navy (Applause). It is needless to say, I hope, that the Committee on Awards did not know that they had bagged such big game. The Society is to be congratulated on having such contributors to its proceedings. The paper by Naval Constructor Taylor will now be read by the author.

Naval Constructor D. W. Taylor read his paper entitled "The Theoretical and Practical Methods of Balancing Marine Engines."

THE THEORETICAL AND PRACTICAL METHODS OF BALANCING MARINE ENGINES.

By Naval Constructor D. W. Taylor, U. S. N., Member of Council.

[Read at the ninth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 14 and 15, 1901.]

SOURCES OF VIBRATIONS OTHER THAN THE ENGINES.

1. Inasmuch as the principal object of balancing marine engines is to avoid the vibrations set up in ships through the operation of unbalanced engines, it may be as well, before taking up in detail the engines, to point out that there are other factors to be considered in connection with the vibration of ships. A propeller may be well balanced and true, but if there is not a clean run of water to it, or, in other words, if the relative velocity of the water with reference to the ship is not uniform throughout the propeller disk, the forces acting upon each blade vary during a revolution. If, as a result of the varying forces upon the blades, there is a variation of the total force on the propeller during the revolution, we have resulting an unbalanced force, acting at the extreme after end of the ship, where it is most favorably situated to cause the latter to vibrate. Moreover the propeller itself may be out of balance or may have one or more blades out of true.

CONCLUSIONS FROM YARROW'S EXPERIMENTS.

2. Some years ago, Mr. Yarrow made some experiments on torpedo-boat vibrations, in the course of which he ran a boat under way and determined the nature and extent of the vibration. He then removed the propeller and ran the boat alongside the dock, determining again the nature and extent of the vibration. The result was that, for a given number of revolutions of the engines, there was very little difference between the vibration under way and the vibration alongside the dock with the propeller removed. The correct inference was that in this case the propeller had little or no tendency to cause vibration. This, however, was for a torpedo-boat with fine lines, a fine run, and a very carefully balanced propeller. It does not follow that for ships

which do not possess these characteristics the propeller has no effect upon vibration. That marked vibration *can* be caused by the propeller is a matter of common experience. Most of us have probably noticed at some time, in the case of a ship coming alongside a wharf, that when the engines are reversed marked vibration is very apt to make its appearance.

SOME ELEMENTARY FEATURES OF VIBRATIONS OF SHIPS.

3. The question of the vibration of ships is beyond the scope of this paper; but in view of the large amount of matter which has been published upon the question within the last fifteen years or so, covering the question of ships' vibrations, as well as engine balancing, it is supposed that the members of the Society are thoroughly familiar with the general phenomena.

Plate 17 shows in an exaggerated way the general nature of common vertical vibrations in ships. The full lines show the ship at rest or in the middle of a vibration, and the dotted lines show the extreme up and down positions. The points *NN* which neither rise nor fall during vibration are called the quiescent points, or, preferably, nodes, and the points *LL* which have the maximum motion are called loops. There must be at least two nodes. Theoretically, there may be any number distributed along the ship with corresponding loops. The more nodes and loops the more rapid the vibrations or the higher their frequency. There is probably a limit to the frequency with which actual ships can vibrate, but there is little positively known on this point.

PRESENT STATUS OF ENGINE BALANCING.

4. The matter of marine engine balancing has been the subject of much study and experimental investigation by distinguished engineers of every shipbuilding country, and so much has been written about it that it is practically impossible at this time to put forward anything essentially new, or even to ascribe methods and results to the authors who first brought them forward. Suffice it to say that the state of the science is now such that, given an engine, it is possible to determine fully the unbalanced forces developed from it at any number of revolutions. In theory it is possible to balance perfectly any engine. In practice, however, so many weighty considerations, apart from those of balancing, must be taken account of that the design of correctly and satisfactorily balanced engines, which are satisfactory in other respects, is exceedingly difficult.

EFFECT OF SLACK BEARINGS.

5. In what I have to say I shall assume there is no slackness of bearings, so that the engine to be balanced is free from pounding or jar. An engine which is pounding because of slackness of bearings frequently causes, through its pounding, tremors throughout the ship, which are often confused with vibrations proper, and are, indeed, almost equally objectionable. Pounding can frequently be reduced, or gotten rid of entirely, through adjusting the compression by proper valve setting. For this reason we sometimes find engineers under the impression that vibrations proper, such as shown in Plate 17, can be reduced or avoided by proper valve setting. This, however, is not the case, as general vibrations are affected only by the unbalanced forces due to moving weights or the propeller action.

GENESIS OF UNBALANCED FORCES.

6. Let us consider first an engine with a single cylinder. If steam is in the cylinder, but the piston is not moving, the pressure upon the piston and the pressure upon the cylinder-head are equal and opposite. These pressures neutralize one another through the engine framing, and there is no external or unbalanced force communicated to the engine supports. If, however, the piston, with its attached weights, is moving and being accelerated, that portion of the steam pressure necessary to accelerate it is absorbed in giving energy to the moving parts, and hence is not transmitted to the engine framing. It follows that the pressure on the cylinder head which tends to move the framing in one direction is only partially balanced by that portion of the steam pressure upon the piston which is transmitted to the engine framing via the piston rod, connecting rod, and crank shaft. It is obvious, then, that the unbalanced forces devoloped when a single-cylinder engine is revolving are equal and opposite to the forces necessary to accelerate or retard the moving parts as the engine turns over.

RESULTS OF INVESTIGATION OF SINGLE-CYLINDER ENGINE.

7. In Appendix A will be found a detailed investigation of the geometry and mechanics of the motion in the case of a single-cylinder engine. Referring to the results in Appendix A, we find that the weights of the moving parts of a single-cylinder direct-acting engine are equivalent to a reciprocating weight W, which moves with the crosshead and a revolving weight L, which is concentrated at the crank-pin axis.

That if r denotes the crank-arm radius (equal to one-half the stroke)

and θ the angle made by the crank radius with the direction of the cylinder axis, we have for a vertical engine an unbalanced force denoted by Q, due to the reciprocating weight W, and unbalanced vertical and horizontal forces denoted by V and H, due to the revolving weight L. If R denotes revolutions per minute, the expressions for Q, V, and H are as follows:—

$$Q = .000341\, rR^2 W \left[\cos\theta + \frac{\cos 2\theta}{n}\right]$$

$$V = .000341\, rR^2 L \cos\theta$$

$$H = .000341\, rR^2 L \sin\theta.$$

FORMULÆ FOR FORCES AND MOMENTS IN MULTI-CYLINDER ENGINES.

8. Suppose, now, we have k parallel cylinders of equal stroke directly connected to the same shaft. Suppose that their axes are distant l_1, l_2, l_3, ... l_k feet, respectively, from a fixed point on the shaft. Suppose their cranks are set at angles a_1, a_2, a_3, ... a_k with a fixed radius of the shaft. Let w_1, w_2, w_3 ... w_k denote the reciprocating weights, L_1, L_2, L_3 ... L_k the revolving weights, r the common crank arm, R the common revolutions per minute, and θ the angle made by the fixed shaft radius (from which a_1, a_2, etc., are measured) with the plane through the axes of the cylinders.

We need to determine the total or resultant unbalanced forces set up because of the rotation of the engine. Since the unbalanced forces in the various cylinders are not all in the same line, unbalanced moments will also make their appearance, and these, too, we need to determine.

Consider, first, the reciprocating weights. Denote the unbalanced forces due to them in the respective cylinders by q_1, q_2 ... q_k. Evidently from the results demonstrated in Appendix A we have:—

$$q_1 = .000341\, rR^2 w_1 \left\{\cos(\theta + a_1) + \frac{1}{n}\cos 2(\theta + a_1)\right\}.$$

Similarly we have

$$q_2 = .000341\, rR^2 w_2 \left\{\cos(\theta + a_2) + \frac{1}{n}\cos 2(\theta + a_2)\right\},$$

and so on. It is convenient in determining the complete effect of the forces to use the ordinary device of applying, at the point through which moments are taken, two equal, opposite, and parallel forces for each unbalanced force considered. Each unbalanced force is thus reduced to a force acting through

the point about which moments are taken, and a couple whose arm is the distance from the point about which moments are taken to the line of action of the unbalanced force. If we take moments about O and denote the moment of the couple, due to w_1 by m_1 we have

$$m_1 = q_1 l_1 = .000341\, rR^2 w_1 l_1 \left\{ \cos(\theta + a_1) + \frac{1}{n} \cos 2(\theta + a_1) \right\}$$

$$m_2 = q_2 l_2 = .000341\, rR^2 w_2 l_2 \left\{ \cos(\theta + a_2) + \frac{1}{n} \cos 2(\theta + a_2) \right\},$$

and so on.

The expression $.000341\, rR^2$ common to all our formulæ may be conveniently replaced by C.

It is also convenient at this stage of analysis to separate the forces into two parts. That part varying as $\cos(\theta+a)$ is conveniently called the primary force, and that varying as $\cos 2(\theta + a)$ is conveniently called the secondary force. Corresponding to the primary and secondary forces we have similarly the primary and secondary moments. Let Q_1 denote the resultant primary force due to all the cylinders. Let Q_2 denote the resultant secondary force due to all the cylinders. Let M_1 denote the moment of resultant primary couple due to all the cylinders. Let M_2 denote the moment of the resultant secondary couple due to all the cylinders. The forces being considered are all parallel, being those due to reciprocating weights, and their resultants are readily written down from well-known principles of mechanics.

Evidently

$$Q_1 = C\,[w_1 \cos(\theta + a_1) + w_2 \cos(\theta + a_2) + \ldots + w_k \cos(\theta + a_k)]$$

$$Q_2 = \frac{C}{n}\,[w_1 \cos 2(\theta + a_1) + w_2 \cos 2(\theta + a_2) + \ldots + w_k \cos 2(\theta + a_k)]$$

$$M_1 = C\,[w_1 l_1 \cos(\theta + a_1) + w_2 l_2 \cos(\theta + a_2) + \ldots + w_k l_k \cos(\theta + a_k)]$$

$$M_2 = \frac{C}{n}\,[w_1 l_1 \cos 2(\theta + a_1) + w_2 l_2 \cos 2(\theta + a_2) + \ldots + w_k l_k \cos 2(\theta + a_k)]\,.$$

INTRODUCTION OF GRAPHIC METHODS.

9. From the above somewhat formidable looking expressions it is possible to calculate the resultant unbalanced forces and moments of unbalanced couples due to any known system of cylinders at any point of the revolution of the shaft. These expressions, however, are entirely too

complicated to be of much value when the investigation of measures of amelioration is undertaken. Fortunately, the case is one to which graphic methods are peculiarly adapted.

Referring to Fig. 4, Plate 18. Ox is the direction of the cylinder axes $xOA = \theta$. Draw OC_1 parallel to crank No. 1 and such that to a suitable scale $OC_1 = w_1$. Similarly draw OC_2, OC_3, OC_4, representing in direction and length the other cranks and reciprocating weights. Four cylinders are represented in the figure, but the process is applicable to any number.

Draw the perpendiculars C_1Q_1, C_2Q_2, C_3Q_3, C_4Q_4.

Evidently $$OQ_1 = w_1 \cos(\theta + \alpha_1)$$

$$OQ_2 = w_2 \cos(\theta + \alpha_2), \text{ and so on.}$$

The algebraic sum of OQ_1, $OQ_2 \ldots OQ_4$ is OQ, which then represents in magnitude and direction the resultant primary force.

Referring now to Fig. 5, Plate 18, instead of drawing all the cranks from O draw Ox and OC_1 as before. Then from C_1 draw a line C_1C_2 parallel to the second crank and representing in length its reciprocating weight. Draw from C_2 a line C_2C_3 similarly representing the third crank, and from C_3 draw C_3C_4 for the fourth crank. Drop the perpendicular C_4Q upon Ox, and OQ in Fig. 5 is exactly the same as OQ in Fig. 4. $OC_1C_2C_4$ in Fig. 5 is called the primary force polygon, and the angle β or AOC_4 is constant, whatever the value of θ. Evidently, if for OC_1, OC_2, and so on, we substitute a single crank at the angle β with OA, driving a reciprocating weight, represented by the length of OC_4, then, as regards primary reciprocating forces, the single weight on the single crank is equivalent of the system of four weights on four cranks from which it was deduced. The line OC_4 is called the closing line of the primary force polygon. Clearly, if there were a fifth crank in the combination, parallel to C_4O and driving a reciprocating weight represented by the length C_4O, there would be no resultant primary force at any point of the shaft's revolution. In other words, the engine would be perfectly balanced as regards primary force.

In practice, when drawing a primary force polygon, it is convenient to make OA correspond with a crank radius, and measure all other crank angles in the direction of rotation from the direction of this crank. I usually use the forward crank radius as the direction to which to refer all angles.

Exactly in the same manner as the primary force polygon has been determined, it is possible to determine and delineate a primary couple or moment polygon, and the secondary force and couple polygons.

In drawing the secondary force polygon the weights are, of course, $\frac{1}{n}$, those used in the primary force polygon, and the angles are doubled. This involves drawing angles greater than 360 degrees, which can, of course, be readily done.

When it comes to the couple or moment polygons, the lengths of the sides represent the moments of the couples, or the products $w_1 l_1$, $w_2 l_2$, etc. It is necessary, when considering actual engines, to choose a convenient point from which to measure l_1, l_2, etc., or, in other words, about which to calculate the moments of the couples. I find it convenient to use the point where the after cylinder axis intersects to shaft axis.

ILLUSTRATIVE FORCE AND MOMENT POLYGONS AND CURVES OF FORCES AND MOMENTS.

10. In Table II will be found the data necessary for the determination of, and in Plate 19 will be found plotted, the primary and secondary force and moment polygons for a 4-cylinder engine.

In order to make the case somewhat clearer, I have shown in Plate 20 the unbalanced forces and couples of the engine of Table II plotted as ordinates above angles of forward crank as abscissæ. The primary and secondary forces and moments are shown plotted separately and then combined, the combined diagram representing, of course, the actual force or moment present. The device of separating these forces and moments into their primary and secondary elements is a convenient and useful one, but it should not be overlooked that the two are really always combined into and appearing as one force and one couple, and not as distinct and separate forces and couples acting simultaneously upon the shaft.

TREATMENT OF REVOLVING WEIGHTS.

11. So much for the reciprocating weights. The forces due to them are all parallel to the cylinder axes—vertical in the case of a vertical engine and horizontal in the case of a horizontal engine. When, however, we consider revolving weights, we find from Appendix A that, for a single weight L, the forces V and H, parallel and perpendicular, respectively, to the cylinder axis, are given by the formulæ—

$$V = CL \cos \theta$$
$$H = CL \sin \theta.$$

Where, as before, $C = .000341\ rR^2$.

It is to be noted that these revolving forces are altogether primary. When proceeding from the case of a single cylinder, fully discussed in Appendix A, to that with a number of cylinders acting on one shaft, the steps in connection with revolving weights are exactly the same as for reciprocating weights, and lead us to force and moment polygons for the revolving weights, both of which are primary only. It is to be noted that the closing line of a force polygon for revolving weights represents a revolving weight. Table III gives data for determining the revolving force and moment polygons for the engine of Table II. In actual engines the revolving weights on each crank are very apt to be equal, but, for the purpose of illustration, they have been assumed different in Table III. The revolving weights are, of course, the virtual weights due to crank pin, crank arms reduced, and that part of the connecting rod which is to be regarded as concentrated at the crank-pin center, etc.

Plate 21 shows the force and moment polygons due to the revolving weights of Table III.

TREATMENT OF MINOR OR SECONDARY WEIGHTS.

12. I have now shown how to determine completely the resultant unbalanced forces and moments, at any point in the line of the shaft, due to any number of cylinders, their pistons, connecting rods, etc.—what may be called the main weights of the engine. There are, however, other moving weights which it is necessary to take account of—such as the valves, air pump gear, bilge pump gear, when fitted, etc.

In Appendix B, I have described and shown in detail methods of handling typical weights—valve gear, etc.—and it is believed that a careful study of the methods applied in Appendix B will enable the student to determine proper results in any ordinary case. There is no satisfactory method for handling radial valve gears, because such gears do not give harmonic motion. The most satisfactory method for such gears is to determine an approximate harmonic motion for them and work with that. The exact method is, of course, to determine graphically the motion of the valve, and then its velocity and acceleration; but even after this is done we are unable to satisfactorily balance such gears with weights which have harmonic motion.

LENGTHS USED FOR SIDES OF SECONDARY POLYGONS.

13. It is convenient, as a rule, to draw the secondary force and moment polygons with sides the same length as those of the primary polygons,

remembering always that the scales for the secondary polygons are $n$times larger than the scales for the corresponding primary polygons.

REASONS FOR SEPARATING FORCES AND MOMENTS.

14. The above completes the description and demonstration of methods for the determination of unbalanced forces and moments of unbalanced couples in a known engine. The unbalanced force acts through the point about which moments are taken. In theory the unbalanced force and couple can always be reduced to a single force acting at some other point of the shaft axis. But as this theoretical point shifts along the axis (unless the closing lines of force and moment polygons are parallel) as the shaft revolves, and is hence generally outside the engine framing, it is not necessary, as a rule, to determine its position, it being really simpler to consider always the force acting at and the moment about one point suitably chosen.

CONDITIONS ESSENTIAL TO BALANCE.

15. I come now to the question of balancing engines so as to reduce or extinguish entirely unbalanced forces. The methods previously explained will always enable us to determine whether or not an engine is balanced, and modifications of them will be found very useful when searching for methods to obtain a balance. I shall consider, primarily, the usual type of marine engine, the "vertical inverted," where the cylinders are arranged in line above the shaft. There are four conditions which must be fulfilled in a perfectly balanced engine.

First. As it revolves the combined center of gravity of the moving reciprocating weights must remain at a constant distance from the axis. When this is the case, there can obviously be no resultant inertia forces, and hence no external force caused by the motion of the reciprocating weights.

Second. If we substitute for each moving reciprocating weight another one proportional to the product of the weight by its distance parallel to the shaft from a fixed point on the shaft axis, the center of gravity of this derived system of weights must remain at a constant distance from the axis as the shaft revolves.

When this is the case and the first condition is complied with, there can be no external couple due to reciprocating weights.

Third. The center of gravity of all revolving weights must be in the shaft axis.

When this is the case, there can be no external force due to revolving weights.

Fourth. If for each revolving weight we substitute a weight proportional to the revolving weight, multiplied by its distance parallel to the shaft from a fixed point on the shaft axis, the center of gravity of this derived system must be in the shaft axis.

When this is the case and the third condition is complied with, there can be no couple or moment from revolving weights.

The above essential conditions for perfect balance have been known for years. It is evident that if all the force and moment polygons, as explained by me, are closed, the four conditions are fulfilled—otherwise, they are not fulfilled. It is evident, too, that if the four conditions are fulfilled for uniform rotation, they are necessarily fulfilled for non-uniform rotation.

OBJECTIONS TO BALANCING RECIPROCATING WEIGHTS BY REVOLVING WEIGHTS.

16. It will be noted that I have throughout made a marked distinction between the reciprocating and the revolving weights. This is because forces and moments due to reciprocating weights are constant in direction, remaining always in the plane of the cylinder axis, and vary in magnitude as the shaft revolves.

The forces and moments due to revolving weights, on the other hand, are constant in amount, but vary in direction as the shaft revolves. The components of the revolving forces and moments, resolved parallel to any one direction (say the plane of the cylinder axes), are variable in amount, because the constant forces and moments are variable in direction.

It is quite possible to balance the forces and moments of reciprocating weights by the components parallel to the plane of the cylinder axes due to revolving weights, but this renders it impossible to balance the forces and moments of the revolving weights perpendicular to the plane of the cylinder axes. If reciprocating weights are balanced by revolving weights, the latter must have unbalanced resultant components in the plane of the cylinder axes equal and opposite to the unbalanced resultants of the reciprocating weights. But then the unbalanced resultants of the revolving weights in the direction perpendicular to the plane of the cylinder axes are equally great, and in this plane there is nothing to oppose or balance them.

Cases are supposable where we would be warranted in trying the effect of balancing reciprocating weights by revolving weights. Suppose, for

instance, serious vertical vibrations occur at the revolutions at which it is most desirable that the engines should run, and are evidently due to synchronism between the revolutions of the engine and the natural period of vibration of the ship in a vertical plane.

We would be warranted in using revolving weights to produce vertical balance, on *the chance* that the unbalanced horizontal forces produced would not synchronize with the period of horizontal vibrations of the ship, and that thus beneficial results would be produced. Revolving weights, however, producing only primary forces and moments, cannot balance the secondary forces and moments of reciprocating weights. Moreover, in actual unbalanced engines the unbalanced forces and moments from reciprocating weights are nearly always much larger than those from revolving weights. So, to balance by revolving weights the primary forces and moments from the reciprocating weights, we must largely add to the unbalanced revolving forces and moments perpendicular to the plane of the reciprocating weights without being able to reduce at all the secondary forces and moments from the reciprocating weights. It is, then, clearly more logical and satisfactory to attack the problem in detail—balance reciprocating weights by reciprocating weights only, and revolving weights by revolving weights only.

FORCE AND MOMENT POLYGONS FOR ACTUAL NAVAL ENGINES.

17. It is evident from what has gone before, that the more sides to our force and moment polygons the more readily they can be closed by manipulating weights and crank angles. That is to say, the more numerous the cranks of an engine the more readily it can be balanced without using special balance weights, which should be avoided where practicable. Also, for an engine of given power and revolutions per minute, the more numerous the cylinders the lighter the moving parts of the individual cylinders.

Perhaps this can be better appreciated from examination of Plates 25, 26, 27, and 28. These give the primary and secondary force and moment polygons for the reciprocating weights of four United States naval vessels. The particulars needed in determining these polygons are given in Table IV below. The Charleston referred to is the first steel cruiser of that name, which was wrecked a year or two since. The polygons of Plates 25 to 28 are drawn on very different scales for the various vessels. If drawn on the same scale for all four, the differences in actual size would be startling.

After making due allowance for the fact that the engines develop different powers at different revolutions, the ameliorative effect, as regards vibrations, of increase in the number of cylinders is obvious.

RANGE OF DISCUSSION TO BE UNDERTAKEN.

18. I shall proceed now to discuss the question of balancing ordinary vertical inverted engines of 1, 2, 3, 4, and 5 cranks. As a rule, there is one cylinder to each crank, and this is assumed to be the case unless otherwise stated. Main weights only will be considered at first.

ENGINES WITH ONE CRANK.

19. In Appendix A the resultant forces due to the inertia of moving parts of a single-crank engine are fully determined. With one cylinder there are no unbalanced moments, and partial or complete balance cannot be obtained by judicious arrangement of moving parts of several cylinders. It is possible to balance the primary reciprocating forces by introducing large revolving weights, but the strong objections to this course have already been pointed out. It is, however, practicable to balance completely the revolving weights by weights on the shaft opposite the crank pin, and this should always be done. If it is then required to balance the reciprocating weights, the most practicable method is to use lever driven balance weights, one arrangement being shown diagrammatically in Fig. 17, Plate 29. The introduction of such a weight evidently results in an unbalanced couple. If this must be avoided, two weights may be used, as indicated in Fig. 18, Plate 29. It is to be observed that, in the case of a single-cylinder engine with an air pump driven by a lever from the crosshead, the reciprocating air-pump weights partially balance those of the cylinder, and by making them of the proper weight complete balance may be obtained. But then the air-pump weights would be impossibly heavy. The small use now made of single-cylinder engines for marine purposes renders the question of the balance of such engines one of little practical importance.

ENGINES WITH TWO CRANKS.

20. Typical force and moment diagrams for the reciprocating weights of a two-crank engine are found in Plate 25 for the U. S. S. Charleston.

It is entirely impossible to close either force or moment polygons of two crank engines without introducing other sides, *i. e.*, with the two-cylinder engine as usually arranged bob weights must be used to improve the balance. But these must be driven by cranks or their equivalents—eccentrics—resulting in an engine with more than two cranks.

It is probably the best practice, if two-crank engines must be balanced, to secure revolving balance by balance weights opposite each crank, and

then for reciprocating balance use lever balance weights for each cylinder, as in the case of a single-cylinder engine. There are special arrangements, however, for *two-cylinder* engines, which secure perfect balance, though the usual arrangement of two cranks at 90 degrees must then be departed from.

Fig. 19, Plate 29, shows a perfectly balanced two-cylinder arrangement. The cranks are 180 degrees apart, and the cylinders have their axes in line and on opposite sides of the shaft. It is necessary in practice to use a double connecting rod and double crank for one cylinder, these being in each case of the same total weight as the corresponding parts for the other cylinder. Of course, too, the reciprocating parts of each cylinder must be of the same weight. It is obvious on inspection that this engine fulfills the four conditions for balance as regards forces set forth in paragraph 15, since, as the shaft revolves, the center of gravity of the moving weights does not change. Also, since the two cylinders are in line, there are no moments developed. The arrangement of Fig. 19 is not applicable to marine engines. It is used for fan and automobile engines. To save expense, steam supply complications, etc., the arrangement shown in Fig. 20, Plate 29, is often adopted, even where that of Fig. 19 could be introduced, and it is used for small marine engines where Fig 19 is not applicable. Fig. 20 is often considered equivalent to Fig. 19, but such is far from being the case. A couple is introduced, since the two cylinder axes are not in the same line, and the arrangement is the worst possible for secondary forces. Consider an engine where the connecting rod is four cranks long, for instance. The primary reciprocating force is abolished if the moving parts are made of the same weight, but, as the secondary forces are added to one another, we have an unbalanced secondary force equal in amount to that due to half of one reciprocating weight.

When we consider revolving forces, the arrangement of Fig. 20 is objectionable only in the introduction of an unbalanced moment. The primary forces are neutralized as for the reciprocating weights, and there are no secondary forces from revolving weights. If the most complete revolving weight balance is wanted, however, the revolving weights for each crank must be separately counterbalanced.

The arrangement of Fig. 20 is a decided improvement, as regards vibrations, over the customary arrangement where the cranks are at 90 degrees. But the latter is preferable from certain practical considerations; for instance, it has no dead center, and gives a much more uniform turning moment.

MACALPINE'S TWO-CYLINDER BALANCE.

21. A two-cylinder arrangement has recently been brought forward by Macalpine, which is superior to that of Fig. 20 in that it eliminates secondary forces. It is shown diagrammatically in Fig. 21, Plate 29. It gives perfect balance as regards reciprocating weights, and its unbalanced couples are athwartships instead of fore and aft, and are not likely to cause objectionable vibration. This arrangement has the practical objections mentioned above to that of Fig. 20, and a few others of its own, due to increase in number of working parts, use of rocking levers, etc. It is, however, a simple and elegant solution of the problem of complete two-cylinder balance, and by doubling it, *i. e.*, using two pairs of cylinders on one shaft, it can be applied to triple and quadruple expansion engines.

ENGINES WITH THREE CRANKS.

22. For three-crank engines the force polygons are three-sided figures and the moment polygons two-sided figures. (See Plate 26, giving the polygons for the Yorktown three-crank engines.)

We have now reached a sufficient number of cranks to allow reduction of unbalanced forces by properly choosing reciprocating weights and crank angles. Also, in these days of triple expansion, all marine engines of importance have at least three cylinders, so the three-crank case is the first one I have discussed of practical importance.

The moment polygons—two-sided figures—obviously cannot be closed by variation of weight and crank angles. The force polygons are, however, three-sided figures, and as long as any two of the reciprocating weights are together greater than the third, the primary force polygon can be made a closed triangle by adopting proper crank angles. The condition that two weights must be greater than the third allows a wide choice of reciprocating weights and crank angles, and we should make a choice which will, if possible—

1. Secure balance of secondary as well as primary forces.
2. Be desirable, or at least not objectionable, as regards the very many considerations other than vibration affecting engine design.

Now if all the reciprocating weights are made equal, and the three cranks set at 120 degrees, we find that—

1. The primary force polygon is a closed equilateral triangle.
2. The secondary force polygon is also a closed equilateral triangle.

3. The crank angles are those fixed upon by common consent as the most desirable from the many considerations other than those of balancing.

Plate 30 shows what the polygons of the Yorktown (Plate 26) would become if the reciprocating weights of each cylinder were made the same as those of the *LP.* It is seen upon comparing it with Plate 26 that primary and secondary forces are abolished, and primary and secondary moments are not seriously increased.

If it is desired to secure reciprocating weight balance as regards moments for three-crank engines, the most feasible plan is to introduce reciprocating balance weights at the ends of the engine driven by cranks or eccentrics. Such an arrangement, however, is in the four or five-crank class.

Considering now revolving weights, it is obvious that if the revolving weight on each crank is equal (cranks being at 120 degrees), forces are completely balanced, while there is left an unbalanced revolving moment, which is, from the nature of the case, smaller than the unbalanced reciprocating moment. For this reason, it is not necessary as a rule to attempt further revolving balance in a three-crank engine with cranks at 120 degrees and equal revolving weights on the cranks, which is, by the way, the usual arrangement, all three cranks and connecting rods being made the same. If complete revolving balance is aimed at, however, for moments as well as forces, the most feasible method is to counterbalance separately the revolving weight on each crank. Or the methods of revolving balance described in discussing four-crank engines may be applied.

EFFECT OF LOCATION WITH RESPECT TO NODES OF THREE-CRANK ENGINE.

23. There is a theoretically practicable arrangement of the three-crank engine, which, though unbalanced as regards both forces and moments, if properly located in the ship, is nearly balanced as regards vibration.

Plate 31 shows the reciprocating force and moment polygons of the Yorktown engines after the increase in L. P. weights from 3,000 to 3,450 pounds, as noted in the legend of the figure. Considering first the primary polygons, it is seen that there is a resultant unbalanced force equivalent to a reciprocating weight of 1,500 pounds on a crank at 133½ degrees. There is also an unbalanced couple with moment of 15,000 pounds feet on a crank at 313½ degrees or 180 degrees from the unbalanced force. The force is acting through and moment taken about the center of the after or L. P. crank. Now, if this point is 10 feet forward of a nodal point (see Plate 17),

the unbalanced force produces at the node a moment of 15,000 pounds feet, equal and opposite to the unbalanced couple, and the net result is no unbalanced moment and a vertical unbalanced force of 1,500 pounds acting through the nodal point where its effect in causing vibrations is at a minimum. Considering the secondary force and moment polygons, the results are just the same. The unbalanced resultant moment is neutralized by the moment of the unbalanced force about the node, and the final result is a virtual unbalanced secondary force at the node where it is least harmful.

BALANCE THAT IS PRACTICABLE WITH THREE CRANKS.

24. The above possibility in connection with three-crank engines is interesting, but of little practical value, owing to the fact that the nodal points of a ship cannot be accurately determined in advance of the ship's completion, and, in fact, are subject to change of location with change of loading the ship. For three-crank engines in practice the nearest approach to balance is obtained by adopting cranks at 120 degrees, and making the reciprocating and revolving weights for each crank the same. A slightly better balance may be obtained by fully balancing with counter-weights the revolving weights for each crank, thus extinguishing revolving moments; but as reciprocating moments, which are usually larger than the revolving moments, are necessarily left unbalanced, it appears hardly worth while, as a rule, to add weight, complication, and expense to a three-crank engine for the purpose of avoiding revolving moments. If a three-crank engine, balanced as regards forces, is placed about the center of the length of the ship, its unbalanced moments are least likely to set up vibrations because unbalanced moments are most harmful where a vibratory node is located, and at the center of the ship there is nearly always a loop. Torpedo-boats with three-crank, equally weighted engines located about the center of length have given fairly satisfactory results as regards vibration.

ENGINES WITH FOUR CRANKS.

25. For a four-crank engine the force polygons are four-sided figures and the moment polygons three-sided figures.

Evidently, then, the "balancing possibilities," so to speak, in the case of a four-crank engine, are decidedly superior to those of engines previously considered. The large and increasing use, of late years, of triple-expansion engines with two L. P. cylinders, and of quadruple expansion engines, has,

through the use of four cranks, increased the opportunities for balancing and the development of balanced types of engines.

Progress in marine engine balance during the last ten years has been made almost entirely with four-crank engines. It is necessary, then, to consider most thoroughly the balancing possibilities and limitations of such engines.

SPACING OF FOUR CRANKS FOR BEST TURNING MOMENT.

26. Before beginning this task, I desire to call attention to the important fact that the usual spacing of cranks of 90 degrees, found in the case of four-crank engines, is essentially inferior as regards uniformity of turning moment.

If a four-crank engine has four simple, double-acting cylinders, the 90 degree spacing is equivalent, as regards turning moment, to two cranks at 90 degrees. For the turning moment on a crank at 0 degrees in such an engine is the same as that on its opposite crank at 180 degrees, and the two cranks at 90 degrees and 270 degrees are equivalent to one crank with double moment at either 90 degrees or 270 degrees. To secure the most uniform turning moment for double-acting engines, the crank should be at or opposite the angles obtained by dividing 180 degrees, not 360 degrees, by 4, the number of cranks. Thus, starting with the first crank at 0 degrees, the second should be at 45 degrees, or 225 degrees; the third at 90 degrees, or 270 degrees; and the fourth at 135 degrees, or 315 degrees. This conclusion, obvious for four-crank simple engines, is readily shown to apply to quadruple engines, where about the same power is developed in each cylinder. We shall see later that the crank angles for four-crank engines, which it is necessary to adopt for balancing, approximate fairly closely to those most favorable for uniform turning, so that a balanced four-crank engine is also one with a more uniform turning moment than if the 90-degree crank spacing is adopted.

It may be asked why, in a three-crank engine, cranks equally spaced around the whole circumference give the best turning moment. The answer is that (since in a double-acting engine the turning moment of a crank is equivalent to that of a crank directly opposite) the usual three-crank arrangement at 0 degrees, 120 degrees, and 240 degrees, is equivalent to one at 0 degrees, 120 degrees, and 60 degrees; and in the latter the cranks are equally spaced over the first 180 degrees of the crank circle.

POSSIBILITIES OF BALANCE WITH FOUR CRANKS.

27. The mathematical investigation of the balancing possibilities in connection with four-crank engines is quite feasible, though laborious. I have attacked the problem by determining first, throughout the practicable range of relative reciprocating weights and cylinder spacings, a large number of arrangements of crank angles, etc., for engines which are completely balanced primarily, and then determined the unbalanced secondary forces and moments for these engines. The details of this work will be found in Appendix C, and in Plate 33 the results are shown graphically. It will be seen that in each little rectangle of this figure is indicated diagrammatically an arrangement fully balanced as regards primary forces and moments. The lines marked 1, 2, 3, and 4 indicate by their lengths the reciprocating weights for the respective cylinders (No. 1 being the after and No. 4 the forward cylinder), and by their direction the corresponding crank angles. The spacings of the cylinders, expressed as fractions of the total length of the engine, are indicated around the margin of the figure. The heavy, full line shows in each case the amount of the resultant secondary force (for the combination shown with connecting rod four cranks long) and the direction of the imaginary crank, with reciprocating weight developing secondary force alone, which corresponds to the resultant. Similarly, the heavy dotted lines determine the moment and direction of the resultant secondary couple. Where a rectangle is left blank it indicates that primary balance is impossible with the assumed weights and cylinder spacing.

From a study of Plate 33, which covers the whole field of four-crank balance, we can readily determine that portion of the field worthy of final detailed investigation.

It will be observed in the first place, that where the crank angles are such as to give an evidently inferior turning moment, the resultant secondary force is usually large, and the relative reciprocating weights such as would be very difficult to obtain in practice.

In the second place, for arrangements where w_2 and w_3 range from 1 to 1.5, the secondary forces are not large, the crank angles approach reasonably close to those which we have seen give the most uniform turning moment, and w_1 does not differ greatly from w_4, which is assumed always as constant and equal to unity.

In the third place, the secondary force varies greatly, both in direction and magnitude, while the secondary moment varies comparatively little, indicating that while we may expect, by further manipulation of weights and

angles, to eliminate secondary force, it will not be possible in practice to eliminate thus the secondary moment.

The obvious inference from the results shown in Plate 33 is that we need examine in detail only arrangements where cylinder No. 4, having unit reciprocating weight and zero crank angle:—

Cylinder No. 3 has reciprocating weight somewhere in the neighborhood of 1.5, and its crank angle in the second quadrant, ranging approximately from 135 to 160 degrees.

Cylinder No. 2 has reciprocating weight somewhere between 1 and 2, and its crank angle in the third quadrant and somewhat less than 270 degrees.

Cylinder No. 1 has reciprocating weight somewhere in the neighborhood of 1, and crank angle in the first quadrant, ranging, approximately, from 45 to 60 degrees.

MOST COMPLETE BALANCE ATTAINABLE WITH FOUR CRANKS.

28. As it is not to be expected that we will be able to get rid of both secondary force and secondary moment by further manipulation of weights and crank angles for four cranks, the next step in my investigation will be directed toward the determining of combinations where secondary forces, as well as primary forces and moments, are extinguished, and the resulting unbalanced secondary moment known.

In Appendix C will be found detailed the graphic methods followed, and their results are shown in Diagrams A to G, Plates 36 to 42, which enable us to determine absolutely the crank angles and relative reciprocating weights for a four-crank engine, with any practicable cylinder spacing, for which primary and secondary forces and primary moments are extinguished. Such engines are upon the Yarrow, Schlick, and Tweedy system, and have been much used abroad, it is claimed, with great success. It is seen that the cylinder axes once located, the crank angles and relative weights follow, and are readily determined by simple inspection from Diagrams A to F. While theoretically any order for the cylinders may be chosen, in practice, owing to the fact that the lightest weights must be on the end cranks, and that uniform turning moments are desirable, it is advisable to adopt the general arrangements outlined below.

For four-cylinder triple-expansion engines the two L. P. cylinders should be outside, and their moving part should be as light as possible. Each L. P. cylinder does only about half the work of the H. P. or M. P., and it is desirable to have the L. P. connecting rods, crank pins, etc., reduced in due proportion.

For four cylinder quadruple expansion engines the H. P. and $M_1P.$ cylinders should be outside and the $M_2P.$ and L. P. inside. Care must be taken to design the L. P. piston as light as practicable, and it is advisable to have the air pump, if worked by a lever, driven from the L. P. crosshead, as thus the L. P. reciprocating parts are virtually lightened.

It is advisable to have the forward pair of cylinders as close together as practicable, and the after pair also as close together as practicable. This result is facilitated if the valves for the two outer cylinders H. P. and $M_1P.$ are placed at the ends of the engine, and the valves for the two inner cylinders $M_2P.$ and L. P. in the middle.

Balance can be had without the precautions noted above, but with much greater additions of weight than would otherwise be needed.

REVOLVING BALANCE FOR FOUR CRANKS.

29. With a four-crank engine, balanced as regards the reciprocating weights, there are so many ways of obtaining revolving balance that it becomes a question of selecting that one having most practical advantages as regards simplicity and cheapness. The ideal method as regards balance alone is to fit proper counter-balance weights opposite each crank. If, however, the revolving force and moment polygons are plotted, it will be found that equally complete balance can be secured with less total addition of weight. If moments are taken about the after crank as usual, a revolving weight at the forward crank at a suitable angle will close the revolving moment polygon. This will, of course, mean an additional side to the revolving force polygon, but the latter can now be closed by a suitable weight at the proper angle at the after crank. In practice it is not convenient, as a rule, to apply balance weights at the cranks, unless they are on or opposite to the crank. So the most feasible plan in practice is to fit two revolving counter-balance weights, one aft of the engine framing and the other forward. The turning wheel can usually be made use of for the attachment of one of these weights, and if the forward end of the shaft is extended two or three inches, a wheel or disk for a forward balance weight is readily attached. Once the correct force and moment polygons are plotted, the amounts and angles of these weights are readily determined.

Before finally balancing revolving weights it is well to take advantage of any obvious methods for reducing the unbalanced resultants. This may often be done, for instance, by drilling holes in one or more solid crank pins, or filling one or more, when they are hollow, with lead or similar devices.

The force and moment polygons will always show what is wanted and indicate the direction along which to work.

ENGINES WITH FIVE CRANKS.

30. It is not necessary to enter into a detailed analysis of the case of five-crank engines. With them perfect balance can be secured in a very simple manner.

We have seen that with three-crank engines, with cranks equally weighted and spaced, both primary and secondary forces are balanced. The balance, as regards forces, is even more complete than this. I have, throughout, taken the acceleration of reciprocating parts as proportional to

$$\cos\theta + \frac{\cos 2\theta}{n}.$$

Macalpine has shown that an even closer approximation is given by taking the acceleration as proportional to

$$\cos\theta + \cos 2\theta\left[\frac{1}{n} + \frac{1}{4}\cdot\frac{1}{n^3} + \frac{15}{128n^5} + \dots\right]$$

$$-\cos 4\theta\left[\frac{1}{4n^3} + \frac{3}{16n^5} + \dots\right] + \cos 6\theta\left[\frac{9}{128n^5} + \dots\right].$$

Now, with a three-crank engine, with cranks equally weighted and spaced, the resultant force due to terms involving $\cos\theta$, $\cos 2\theta$ and $\cos 4\theta$, is zero, so that we may consider that primary, secondary, and tertiary forces all vanish.

Evidently, then, we should obtain almost an ideal balance with three crank engines, as above, were it not for the fact that the cylinder axes, being not in the same line, moments are produced. Now, with a five-crank engine, we can make an arrangement virtually equivalent to a three-crank with cranks equally weighted and spaced, and the three cylinder axes coincident.

Two parallel cranks are equivalent to one crank with weight on it equal to the sum on the two, and located so that each of the two has the same moment about it.

To secure complete balance, then, with five cranks, it is necessary to arrange the heaviest weight in the middle, make the other cranks two and two parallel, one pair at 120 degrees with the middle crank, and the other pair at 240 degrees. Considering each pair, the moment of the weight on

the forward crank about the middle one must be equal and opposite to the moment of the weight on the after parallel crank. Fig. 28, Plate 34, shows a diagrammatic arrangement with five cranks completely balanced, and Fig. 29 the corresponding primary and secondary force and moment polygons, which are seen to be equilateral triangles, one or more sides corresponding to two cylinders.

There has not been experience enough with the five-crank system to settle the best arrangements, but it is readily adapted to a quadruple engine with the first or second M. P. divided into two. Fig. 28 indicates the H. P. divided into two, but this appears undesirable for practical reasons. The two cranks corresponding to the divided cylinder should not be made parallel, but one should be parallel to the H. P. and one to the other M. P. This is for the purpose of gaining the most uniform turning moment possible. The five-crank completely balanced arrangement is practically as good as any arrangement as regards uniformity of turning moment.

REVOLVING BALANCE FOR FIVE CRANKS.

31. To secure complete revolving balance with the five-crank arrangement is not difficult. As a general thing the revolving weights would naturally be the same for each crank. If weight is added to the middle crank, and the revolving weights on each pair partially counterbalanced until for each pair the total virtual weight is equal to the virtual weight on the middle crank, and each one of the pair has the same moment about the middle crank, perfect revolving balance is secured. It is simpler and cheaper, however, to use two revolving balance weights, spaced as far apart as possible, as in the case of four-crank engines.

BALANCE INCLUDING VALVE GEAR AND OTHER SECONDARY WEIGHTS.

32. An engine might be completely balanced as regards the main moving parts and yet develop very serious unbalanced forces due to its other moving weights, such as valve gears, air pumps, etc.

It is impossible to discuss in detail the possible cases which may arise in connection with these. In Appendix B, I have shown how to determine the effects of any type of weight likely to be met with in practice. In balancing, skill and experience in applying the methods I have described are essential, but the modifications necessary to handle valve gears and other secondary weights are very simple in principle. Take an eccentric valve gear, for instance. The eccentrics driving it will be set at certain angles ahead or behind the crank of the cylinder; and the position of the crank once determined, the effect of the valve gear weights can be obtained. So when balanc-

ing a four-crank engine, for instance, first neglect the valve gears, and determine crank angles and main weights. This can be done at once from my Diagrams A to F, Plates 36 to 41, when the spacing of cylinders has been fixed. Then using these crank angles, determine the resultant unbalanced forces due to the valve gears. Next make a second approximation to the main gear weights and angles, so determining them that they are not in perfect balance with each other but balance the valve gears. In rare cases it may be necessary to repeat this process, but as a rule the second approximation to the main crank angles and weights is practically exact. The above is peculiarly applicable to the four-crank engine, where crank angles are varied in balancing. Engines with three or fewer cranks cannot be balanced anyhow. For five-crank engines with cranks 120 degrees apart, the valve gears must be balanced independently like the main gears. This is very simple, if the valves all supply steam on the outside or all on the inside. Then the eccentrics all have practically the same angular advance, and balancing is a question of weighting, which is easily arranged for.

Revolving secondary weights are easily combined with the main revolving weights, and if two revolving balance weights, spaced as far apart as possible, are used, they can readily be arranged to balance all revolving weights, main and secondary. One great advantage of this method of securing revolving balance is, that with provision made for the two balance weights, their exact amounts and angles need not be settled until the engine is practically completed and the exact revolving weights to be balanced known.

IMPORTANCE OF DRAWING FORCE AND MOMENT POLYGONS.

33. In conclusion, I would recommend in the strongest manner that when designing an engine, whether for balance or not, complete force and moment polygons be always drawn, representing the effects of *every moving weight*, the weights being estimated. For completed engines, *actual weights* should be obtained and used for the polygons.

If complete or partial balance is aimed at, these polygons, correctly drawn, enable us to determine at a glance how closely the desired results have been obtained. As illustrating this, reference may be made to Plate 28, showing the polygons for the five-crank engines of the Cushing. The force polygons are closed and the moment polygons have material resultants. It is evident that if the angles of the first and third cranks were interchanged the force polygons would still be closed, while the unbalanced moment resultants, both primary and secondary, would be but fractions of what they are in Plate 28.

TABLE I.

Crank angle.	Conn. Rod Ratio $n = 4$.				Conn. Rod Ratio $n = 7$.				Conn. Rod Ratio $n = 10$.			
	Acceleration.		Error of Formula.		Acceleration.		Error of Formula.		Acceleration.		Error of Formula.	
0	Exact.	From approx. formula.	Actual.	% of max. accel.	Exact.	From approx. formula.	Actual.	% of max. accel.	Exact.	From approx. formula.	Actual.	% of max. accel.
0	1.25000	1.25000	0	0	1.14285	1.14285	0	0	1.10000	1.10000	0	0
15	1.18387	1.18243	−.00144	−0.115	1.08991	1.08965	−.00026	−0.023	1.05262	1.05253	−.00009	−0.008
30	.99501	.99102	−.00399	−0.319	.93818	.93745	−.00073	−0.064	.91627	.91602	−.00025	−0.023
45	.71121	.70711	−.00410	−0.328	.70785	.70711	−.00074	−0.065	.70736	.70711	−.00025	−0.023
60	.37511	.37500	−.00011	−0.010	.42858	.42857	−.00001	0	.45000	.45000	0	0
75	.03678	.04232	+.00554	+0.443	.13410	.13510	+.00100	+0.087	.17187	.17222	+.00035	+0.032
90	−.25820	−.25000	+.00820	+0.656	−.14434	−.14285	+.00149	+0.130	−.10050	−.10000	+.00050	+0.046
105	−.48086	−.47532	+.00554	+0.443	−.38354	−.38254	+.00100	+0.087	−.34577	−.34542	+.00035	+0.032
120	−.62489	−.62500	−.00011	−0.010	−.57142	−.57143	−.00001	0	−.55000	−.55000	0	0
135	−.70301	−.70711	−.00410	−0.328	−.70637	−.70711	−.00074	−.065	−.70686	−.70711	−.00025	−0.023
150	−.73703	−.74102	−.00399	−0.319	−.79386	−.79459	−.00073	−0.064	−.81577	−.81602	−.00025	−0.023
165	−.74799	−.74943	−.00144	−0.115	−.84195	−.84221	−.00026	−0.023	−.87924	−.87833	−.00009	−0.008
180	−.75000	−.75000	0	0	−.85715	−.85715	0	0	−.90000	−.90000	0	0

Values from 180° to 360° simply repeat in opposite order those from 0° to 180°.

The above table shows the error resulting from the assumption that the acceleration of a piston at any point is proportioned to $\cos\theta + \frac{\cos 2\theta}{n}$ where $\theta°$ is the crank angle and n the connecting rod ratio. In this table the accelerations are expressed by numbers the scale being such that unity represents the maximum acceleration of a piston where the connecting rod is infinitely long.

TABLE II.

	Aft.			Forward.
No. of cylinder	1	2	3	4
Reciprocating weight, in pounds	760	1080	1440	1840
Crank ahead of No. 4, or forward crank	280°	160°	75°	0°
Leverage about after crank—feet	0	4.00	9.00	14.50
Moment about after crank—lbs.-feet	0	4320	12960	26680

Length of crank arm r in feet 1.25
Revolutions per minute N 160
Value of .000341 rN^2 = ... 10.912
Connecting rod ratio n = ... 4

TABLE III.

	Aft.			Forward.
No. of cylinder	1	2	3	4
Revolving weight, in pounds	620	750	800	900
Crank ahead of No. 4 on forward crank	280°	160°	75°	0°
Leverage about after crank—feet	0	4.00	9.00	14.50
Moment, in lbs.-ft.	0	3000	7200	13050

Length of crank arm r in feet 1.25
Revolutions per minute N 160
Value of .000341 rN^2 = ... 10.912

TABLE IV.

DATA GIVEN FOR ONE ENGINE IN EACH CASE.

	NAME OF VESSEL.														
	Charleston.		Yorktown.			Vesuvius.				Cushing.					
No. of cylinders	2		3			4				5					
Length of stroke—feet	3.0		2.50			1.6667				1.25					
Length of crank arm—feet	1.5		1.25			0.8333				0.625					
Length of conn. rod—feet	6		5.333			3.75				3.125					
Value of N	4		4.267			4.5				5.0					
Max. I. H. P.	3333		1696			1897				860					
Max. revs. per minute	115		160			280				370					
Value of .000341 R^2	6.765		10.912			22.278				29.177					
Cylinder No.	1	2	1	2	3	1	2	3	4	1	2	3	4	5	
Aft or forward	Aft.	Ford.	Aft.		Ford.	Aft.			Ford.	Aft.				Ford.	
Leverage about aft crank—feet	0	6.75	0	5.65	9.25	0	3.5	7.5	11.0	0	3.50	9.00	13.25	17.25	
Crank ahead forward crank—degrees	270	0	120	240	0	90	270	180	0	216	72	288	144	0	
Actual recip. wt.—lbs	10000	6800	3000	2200	1800	1080	1080	1010	860	210	210	210	210	210	
Moment for primary polygon	0	45900	0	12430	16650	0	3780	7575	9460	0	735	1890	2782.5	3622.5	
Angle for secondary polygon	540	0	240	480	0	180	540	360	0	432	144	576	288	0	
Wt. for secondary polygon	2500	1700	703	516	422	240	240	224	191	42	42	42	42	42	
Moment for secondary polygon	0	11475	0	2913	3902	0	840	1683	2102	0	147	378	556.5	724.5	

APPENDIX A.

DISCUSSION OF A SINGLE-CYLINDER ENGINE.

Expressions for position of a point on the connecting rod.

1. In Figure 2, Plate 18:—

O represents the crank shaft axis of a single-cylinder engine.

Ox is the direction of the cylinder axis.

Oy is perpendicular to Ox.

AB represents the connecting rod of length. A is the crosshead end, and B the crank-pin end.

OB represents the crank arm or radius, denoted by r.

Also θ and φ denote as shown the angles which the crank arm and the connecting rod respectively make with Ox.

Suppose the length (c) of the connecting rod AB to be n times that (r) of the crank arm OB. Then $c = nr$. I shall hereafter usually call n the "connecting rod ratio."

As the crank turns, the motion of the piston, piston rod, and any fitting rigidly attached to them is identical with the motion of A, the crosshead end of the connecting rod. If, then, we determine fully the motion of every point on the connecting rod, including that of A, the motions of all moving parts are known.

Denote by P a point (co-ordinates x, y) on the connecting rod. Let $BP - mBA = mc$.

Now $BM = r \sin \theta = c \sin \varphi = nr \sin \varphi$.

Whence

$$\sin \varphi = \frac{1}{n} \sin \theta$$

$$\cos \varphi = \frac{1}{n} \sqrt{n^2 - \sin^2 \theta}.$$

The position of P is determined by its co-ordinates x and y.

We have

$$x = ON = OM + MN = r \cos \theta + mc \cos \varphi = r (\cos \theta + mn \cos \varphi)$$

$$y = PN = AP \sin \varphi = (1 - m) c \sin \varphi = (1 - m) nr \sin \varphi .$$

VELOCITIES OF POINT ON CONNECTING ROD.

2. Substituting for $\cos\varphi$ and $\sin\varphi$ their values as functions of θ and n and reducing, we have

$$x = r(\cos\theta + m\sqrt{n^2 - \sin^2\theta}), \quad y = r(1 - m)\sin\theta .$$

If the crank is turning we can determine expressions for the velocity of P parallel to Ox and Oy by differentiating with respect to time, t, the values of x and y.

The results are

$$\frac{dx}{dt} = -r\left[\sin\theta + \frac{m}{2}\frac{\sin 2\theta}{(n^2 - \sin^2\theta)^{\frac{1}{2}}}\right]\frac{d\theta}{dt}$$

$$\frac{dy}{dt} = r(1 - m)\cos\theta\,\frac{d\theta}{dt}.$$

ACCELERATIONS OF POINT ON CONNECTING ROD.

3. To determine the inertia forces, or, more properly, forces necessary to overcome inertia, acting upon a moving mass, we need to determine not the velocity of the mass but its accelerations, *i. e.*, changes of velocity.

Now the accelerations of P parallel to Ox and Oy are readily obtained by differentiating the velocities with respect to time.

When this is done we have

$$\frac{d}{dt}\frac{dx}{dt} = \frac{d^2x}{dt^2} = -r\left[\sin\theta + \frac{m}{2}\frac{\sin 2\theta}{(n^2 - \sin^2\theta)^{\frac{1}{2}}}\right]\frac{d^2\theta}{dt^2}$$

$$- r\left[\cos\theta + m\,\frac{n^2\cos 2\theta + \sin^4\theta}{(n^2 - \sin^2\theta)^{\frac{3}{2}}}\right]\left[\frac{d\theta}{dt}\right]^2$$

$$\frac{d}{dt}\frac{dy}{dt} = \frac{d^2y}{dt^2} = r(1 - m)\cos\theta\,\frac{d^2\theta}{dt^2} - r(1 - m)\sin\theta\left[\frac{d\theta}{dt}\right]^2.$$

INFLUENCE OF NON-UNIFORM ROTATION.

4. It is seen that each of the above expressions for acceleration has one term into which $\frac{d^2\theta}{dt^2}$ enters as a factor and one into which $\left[\frac{d\theta}{dt}\right]^2$ enters.

Now $\frac{d^2\theta}{dt^2}$ is the acceleration of the angular velocity of a crank which is

not revolving uniformly, and $\left[\frac{d\theta}{dt}\right]^2$ is the square of the instantaneous angular velocity in the case of a shaft not revolving uniformly and of the uniform angular velocity when there is uniform rotation.

It is customary to assume uniform rotation of the shaft, but since absolutely uniform rotation is not physically possible, it is desirable before going further to get some idea how far this assumption is warranted.

Let us denote the average angular velocity in absolute units per second of the shaft during one revolution by ω. In order to apply to our previous results we need to make some assumptions with reference to the variation of speed of rotation. I shall assume the sine law as sufficiently representative, and denote by k the percentage of the mean speed by which the maximum and minimum speeds are in excess or defect of the mean.

Then at any angle

$$\frac{d\theta}{dt} = \omega\,[1 + k \sin(\theta + \alpha)]$$

$$\frac{d^2\theta}{dt^2} = \omega k \cos(\theta + \alpha)\,\frac{d\theta}{dt} = \omega^2 k \cos(\theta + \alpha) + \omega^2 k^2 \cos(\theta + \alpha)\sin(\theta + \alpha)$$

$= \omega^2 k \cos(\theta + \alpha)$ very nearly, since k is small and k^2 may be neglected in comparison with k.

Substituting these expressions for $\frac{d\theta}{dt}$ and $\frac{d^2\theta}{dt^2}$ in the formulæ for the acceleration of P, we have

$$\frac{d^2x}{dt^2} = -\omega^2 r\Big[\{1 + k\sin(\theta + \alpha)\}^2\left\{\cos\theta + m\,\frac{n^2\cos 2\theta + \sin^4\theta}{(n^2 - \sin^2\theta)^{3/2}}\right\} + k\cos(\theta + \alpha)\left\{\sin\theta + \frac{m}{2}\,\frac{\sin 2\theta}{(n^2 - \sin^2\theta)^{1/2}}\right\}\Big]$$

$$\frac{d^2y}{dt^2} = -\omega^2 r\,(1 - m)[\sin\theta\,\{1 + k\sin(\theta + \alpha)\}^2 - k\cos\theta\cos(\theta + \alpha)].$$

In these expressions α may have any value since the maximum acceleration of the crank pin in its path may come at any point in the revolution.

It appears entirely reasonable to assume that in a marine engine requiring balance the speed of revolution would never during one revolution rise above or fall below the mean more than 10 per cent.

Then for this extreme assumption $k = 0.1$.

From an inspection of the above formulæ it is evident that if k does not exceed 0.1 the acceleration of the moving parts due to acceleration of the

crank pin in its path will not exceed ¼ of the maximum acceleration due to absolutely uniform rotation.

It is evident, then, that the unbalanced forces to be expected in practice from non-uniform rotation are not large compared with those due to uniform rotation. They could not, however, be neglected were it not for the fact which will be made evident later, that an engine which is perfectly balanced for uniform rotation is still perfectly balanced if the speed of rotation is not uniform, and the more nearly perfect balance with uniform rotation is approached the more nearly it is approached, *pari passu*, with non-uniform rotation.

So I shall now proceed upon the assumption that $\frac{d^2\theta}{dt^2} = 0$ and $\frac{d\theta}{dt}$ is constant.

SIMPLIFICATION OF EXPRESSIONS FOR ACCELERATION.

5. Write $\frac{d\theta}{dt} = \omega$ and $k = 0$ in the expressions obtained above.

We have upon reducing

$$\frac{d^2x}{dt^2} = -\omega^2 r\{\cos\theta + m\frac{n^2\cos 2\theta + \sin^4\theta}{(n^2 - \sin^2\theta)^{3/2}}$$

$$\frac{d^2y}{dt^2} = -\omega^2 r\,(1 - m)\sin\theta\,.$$

The second term in the expression for acceleration parallel to the cylinder axis is a clumsy one, and it is desirable to replace it by one which is sufficiently accurate and simple.

We do this by writing

$$\frac{d^2x}{dt^2} = -\omega^2 r\left\{\cos\theta + \frac{m}{n}\cos 2\theta\right\}.$$

The maximum error from this simplification occurs when $m = 1$, being zero when $m = 0$.

Table I, appended, compares the exact and approximate values of

$$\frac{d^2x}{dt^2} \div (-\omega^2 r)$$

for various crank angles and for connecting-rod ratios of 4, 7, and 10.

It is seen that the maximum error from the use of the approximate for-

mula falls off rapidly with increase of length of connecting rod, and is in any case but a fraction of one per cent. of the maximum acceleration; that it occurs at a crank angle of 90 degrees; that the error at top and bottom of stroke is nil, and that the average error throughout the revolution is even for an engine with connecting rod twice the length of the stroke, less than one-quarter of one per cent. of the maximum force involved. It is concluded then, that we are justified in adopting the following expressions as adequately representing the motion of any point on the connecting rod.

$$\frac{d^2x}{dt^2} = -\omega^2 r\left\{\cos\theta + \frac{m}{n}\cos 2\theta\right\}$$

$$\frac{d^2y}{dt^2} = -\omega^2 r\,(1-m)\sin\theta.$$

PRIMARY AND SECONDARY ACCELERATIONS.

6. It is seen that the expression for the acceleration parallel to the cylinder axis involves two terms. The second or smaller term vanishes if $n = \infty$, *i. e.*, for an infinitely long connecting rod. The acceleration corresponding to the first term and independent of the length of the connecting rod is conveniently termed the primary acceleration.

The acceleration corresponding to the second term, depending upon the length and obliquity of the connecting rod, is conveniently termed the *secondary* acceleration.

It is seen that perpendicular to the axis of the cylinder there is primary acceleration only.

DYNAMICS OF MOTION OF A POINT ON THE CONNECTING ROD.

7. Having disposed of the geometry of the motion, dynamical considerations may be taken up. Referring to Fig. 3, Plate 18, suppose P is a point of weight w on the connecting rod, constrained to move in its path by the forces acting at A and B upon the rigid connecting rod. Now the force at A, the crosshead, can act only perpendicular to Ox, the direction of the slide.

Denote it by s. Resolve the force at B into two components, q and p, parallel and perpendicular to Ox.

Then from the well-known formulæ of mechanics, g being the acceleration due to gravity,

$$q = -\frac{w}{g}\frac{d^2x}{dt^2} = \frac{w}{g}\omega^2 r\left[\cos\theta + m\frac{\cos 2\theta}{n}\right]$$

$$p + s = -\frac{w}{g}\frac{d^2y}{dt^2} = \frac{w}{g}\omega^2 r\,(1-m)\sin\theta.$$

A third equation from the angular acceleration of P can be obtained so that, theoretically, q, p, and s can each be determined, but the third equation is complicated, and for our purposes p and s need not be separated.

From the expression for q it is seen that the inertia forces upon a connecting rod are peculiar, in that the primary force, corresponding to the primary acceleration, is independent of the position of the point upon the rod, while the secondary force, dependent upon secondary acceleration, is dependent upon the position of the point—or value of m—being a maximum at the crosshead end and vanishing at the crank end.

SUMMATION OF ELEMENTARY FORCES FOR WHOLE CONNECTING ROD.

8. Let us next sum up the elementary forces, first rewriting the value of q as follows:—

$$q = \frac{w}{g}\omega^2 r\,(1 - m)\cos\theta + m\frac{w}{g}\omega^2 r\left[\cos\theta + \frac{\cos 2\theta}{n}\right].$$

Let

$$Q = \Sigma q = \frac{\omega^2 r}{g}\cos\theta\,\Sigma w\,(1 - m) + \frac{\omega^2 r}{g}\left[\cos\theta + \frac{\cos 2\theta}{n}\right]\Sigma mw$$

$$P + S = \Sigma p + \Sigma s = \frac{\omega^2 r}{g}\sin\theta\,\Sigma w\,(1 - m).$$

Now let $\bar{m}$ be the value of m corresponding to the position of the center of gravity of the connecting rod, W the weight of the connecting rod.

Then

$$\Sigma w\,(1 - m) = W(1 - \bar{m})$$

$$\Sigma wm = W\bar{m}.$$

Whence

$$Q = \frac{\omega^2 r}{g}\cos\theta\,W(1 - \bar{m}) + \frac{\omega^2 r}{g}\left[\cos\theta + \frac{\cos 2\theta}{n}\right]W\bar{m}$$

$$P + S = \frac{\omega^2 r}{g}\sin\theta\,W(1 - \bar{m}).$$

But if we take a weight $W(1 - \bar{m})$ and concentrate it at the crank, and a weight $W\bar{m}$ and concentrate it at the crosshead, and then upon the assumption of an imponderable connecting rod rigidly connecting these weights, determine expressions for the inertia reactions of the system, they are clearly identical with the above expressions for the forces set up by the actual ponderable connecting rod. Hence we may regard $(1 - \bar{m})\times$ (weight of rod) as concentrated at the crank and $\bar{m}\times$ (weight of rod) as concentrated at the crosshead.

In other words, we add part of the connecting rod weight to the revolving weights, supposed concentrated at the crank pin center, and part to the reciprocating weights, which move with the crosshead. In the case of an actual connecting rod the weight and center of gravity can be determined by calculation, and hence the proportions in which the weight should be divided.

Macalpine has stated in the Journal of the American Society of Naval Engineers (Vol. IX, p. 517), that in a large number of cases where the connecting rod was four cranks long (the usual man-of-war length) the C. G. was very approximately .6 the length from the crosshead end. This gives in my notation $\overline{m} = .4$. Probably for the majority of connecting rods which are more than four cranks long $\overline{m}$ is greater than .4, approximating though not reaching .5.

The old rule of thumb was to treat the connecting rod weight as if one-half of it rotated with the crank pin, and the other half moved with the piston. It is seen that this is not much in error, the error being in assigning somewhat too much weight to move with the piston and somewhat too little to rotate with the crank pin. A closer approximation and one amply accurate for many cases where the C. G. calculation would be difficult or tedious is to assume that the weight of the top half of the rod is concentrated at the crosshead and of the bottom half at the crank pin. It is a good plan always to weigh and determine experimentally the position of the center of gravity of a completed connecting rod.

FINAL FORMULA FOR RECIPROCATING WEIGHT.

9. We are now in a position to give final formulæ for the unbalanced forces due to acceleration of all moving parts of a single cylinder engine. The piston, piston rod, all parts rigidly attached to them, and a definite fraction of the connecting-rod weight move as the crosshead for which in our general formulæ $m = 1$. The above are conveniently called the reciprocating weights.

The acceleration of these parts is all in the line of the cylinder axis and given by the formula

$$\frac{d^2x}{dt^2} = -\omega^2 r\left[\cos\theta + \frac{\cos 2\theta}{n}\right].$$

If Q denote the unbalanced force, reckoned positive along Ox, and W the total reciprocating weight, then

$$Q = -\frac{W}{g}\frac{d^2x}{dt^2} = \frac{W}{g}\omega^2 r\left[\cos\theta + \frac{\cos 2\theta}{n}\right].$$

If R denote revolutions per minute, $\omega = \frac{2\pi R}{60} = .10472\, R.$

Also, with sufficient approximation, $g = 32.16.$

Whence finally

$$Q = .000341\, rR^2 W \left[\cos\theta + \frac{\cos 2\theta}{n}\right].$$

FINAL FORMULÆ FOR REVOLVING WEIGHT.

10. We have seen that a fraction of the connecting-rod weight may be considered concentrated at the crank-pin center. The crank-pin weight is also taken as concentrated at its center. The crank arms and other non-symmetrical weights attached to the crank shaft had best be reduced to equivalent weights at the crank radius. This is readily done.

Take a revolving weight w' at radius r', on the crank radius.

Then from our general formula the accelerations of such a weight are evidently

$$\frac{d^2x}{dt^2} = -\omega^2 r' \cos\theta \quad \frac{d^2y}{dt^2} = -\omega^2 r' \sin\theta.$$

The resultant unbalanced forces are

$$-\frac{w'}{g}\frac{d^2x}{dt^2} = w'r'\,\frac{\omega^2\cos\theta}{g} \quad \text{and} \quad -\frac{w'}{g}\frac{d^2y}{dt^2} = w'r'\,\frac{\omega^2\sin\theta}{g}.$$

These are exactly equivalent to the unbalanced forces due to a weight w at crank radius r if $wr = w'r'$. Whence w the virtual weight at crank pin center equivalent to w' is equal to $\frac{w'r'}{r}$.

In practice the revolving weights in connection with a crank which are not symmetrical about the shaft axis are nearly always symmetrical on either side of the center line of the crank radius. If not the virtual revolving weight when reduced to the crank radius will have its center ahead of or behind the crank-pin axis.

I shall proceed on the assumption that the virtual revolving weight is concentric with the crank-pin axis. The changes in methods if this weight is not so concentric are simple and will be obvious.

If we denote by L the total virtual revolving weight centered at the crank axis (called briefly the revolving weight) our general formulæ derived for any point on the connecting rod are applicable to it if we put $m = 0$. Doing this we have

$$\frac{d^2x}{dt^2} = -\omega^2 r\cos\theta \quad \frac{d^2y}{dt^2} = -\omega^2 r\sin\theta.$$

If V denote the unbalanced force parallel to x, and H that perpendicular to x, we have

$$V = -\frac{L}{g}\frac{d^2x}{dt^2} = \frac{L}{g}\omega^2 r \cos\theta.$$

$$H = -\frac{L}{g}\frac{d^2y}{dt^2} = \frac{L}{g}\omega^2 r \sin\theta.$$

Substituting for ω its value in terms of revolutions per minute, R, and for g its value of 32.16, we have, finally,

$$V = .000341\ rR^2L \cos\theta.$$

$$H = .000341\ rR^2L \sin\theta.$$

For a vertical direct-acting engine V is positive upward, and θ is measured from the highest position of the crank-pin center.

APPENDIX B.

Discussion of Various Appendages.

GENERAL PRINCIPLES.

1. In addition to the piston, piston rod, crosshead, connecting rod, crank pin, and crank arms, usually called the main gear, there are many moving parts, as a rule smaller and subsidiary, connected with every engine, such as valve gear, direct driven air pumps, etc.

The work of determining the effect of such weights as regards unbalanced forces and moments is, as a rule, more complicated than that of handling the main gear. The principles involved, however, are comparatively simple, and, with a systematic and orderly arrangement of the calculations, there is no difficulty in reaching correct results.

In handling such weights it is advisable always to reduce them as soon as possible to virtual weights, having the travel of the piston for reciprocating weights and revolving with the crank radius for revolving weights. The advantage of this is that one scale can be used for polygons of the secondary weights and main weights, and, in fact, one polygon can be drawn for both main and secondary weights.

In reducing the secondary weights to virtual weights having the same motion as the main weights, there is one point requiring especial care. As a rule, secondary reciprocating weights have not only a different stroke, but a different connecting-rod ratio from the main weights. For such a weight the virtual weight for primary diagrams is obtained very simply by multiplying it by the ratio (in all practical cases a fraction) between the travel of the secondary weight and the stroke of the engine. For the secondary diagrams, however, it is necessary to multiply this equivalent primary weight again, not by the connecting-rod ratio of the main weights, but by the connecting-rod ratio of the secondary weight.

The methods to be followed in dealing with the secondary weights can be best made clear from examples. I shall take three examples which will cover the ground as regards any case likely to occur in practice. They are:—

1. The ordinary double eccentric sliding link valve gear.
2. An air pump, lever driven from a crosshead.
3. A bilge pump, eccentric driven from the main shaft.

VALVE GEAR.

2. The sketch, Fig. 9, Plate 22, which is not drawn to scale, shows an ordinary double eccentric valve gear in the go-ahead position, the data for it being as follows:—

DIMENSIONS.

Stroke of engine	42 inches.
Connecting-rod ratio for engine	4
Travel of valve	6 "
Eccentric rod length from center of sheave to center of top pin	84 "
Steam admitted on outside of valve	
Angular advance of eccentrics	135 degrees.

WEIGHTS.

Weight of valve, balance piston, valve stem and radius link block	1,446 lbs.
Weight of radius links	160 "
Weight of drag links or reversing links	92 "
Total weight of each eccentric rod and strap	480 "
Weight of top half of eccentric rod	125 "
Weight of each eccentric sheave	275 "

It is necessary to separate the ahead gear from the astern gear and the revolving from the reciprocating weights.

The angular advance being 135 degrees, the ahead gear is driven by a crank 135 degrees ahead of the main crank of the cylinder to which the valve belongs, and the astern gear by a crank 135 degrees behind, or preferably $360 - 135 = 225°$ ahead of the main crank. The connecting-rod ratio for the valve gear is $\frac{3}{84} = 28$. The travel of the valve being six inches and the engine stroke forty-two inches, the primary reduction factor for the primary polygon is $\frac{6}{42} = \frac{1}{7}$. Since the connecting-rod ratio for the valve gear is $\frac{3}{84} = 28$, for the secondary polygons the reduction factor becomes $\frac{6}{42} \times \frac{1}{28} = \frac{1}{196}$.

Taking up now the weights, first separate the actual weights into ahead and astern, reciprocating and revolving.

The first item—valve and rigid attachments—is all reciprocating and all belongs to the ahead gear.

The second item is 160 pounds for radius links. The radius links are driven by both the ahead and astern gear. They reciprocate with motion not absolutely harmonic, but it is an ample approximation if we divide the weight equally between the ahead and astern gear, giving 80 pounds to each.

The drag links, or reversing links, weigh 92 pounds. One end is at rest while the other reciprocates with the valve stem, since, as shown in the figure, these links take hold on the go-ahead end of the radius links. So we add 46 pounds to the go-ahead reciprocating weights.

Next we have the eccentric rod and strap. There is one for the ahead gear and one for the astern gear. This weight is like that of a connecting rod (which it is in essence) partly reciprocating and partly revolving, being concentrated at the two ends in the inverse ratio of the distances of its center of gravity from the ends. In practice a very close approximation is obtained by regarding the weight of the top half of the rod as concentrated at the eccentric centre and revolving. So for both ahead and astern gears we have, from the eccentric rods, 125 pounds reciprocating and 355 revolving.

The eccentric sheaves are, of course, all revolving, adding 275 pounds to both ahead and astern revolving weights.

It is necessary now to collect the actual weights and apply the factors of reduction, by which we get the corresponding virtual weights reduced to engine strokes, etc.

The work is conveniently done in tabular form as indicated below :—

	Ahead Gear.			Astern Gear.	
	Total Weight.	Regarded as Recipro-cating.	Regarded as Revolving.	Regarded as Recipro-cating.	Regarded as Revolving.
Valve and rigid attachment	1446	1446	—	—	—
Radius links	160	80	—	80	—
Drag links	92	46	—	—	—
Eccentric rod and strap	480	125	355	125	355
Eccentric sheave	275	—	275	—	275
Totals		1698	630	205	630
Virtual primary weights—Factor $\frac{1}{7}$	—	243	90	29	90
Virtual secondary weights—Factor $\frac{1}{17\frac{1}{2}}$	—	19	—	1	—
Crank ahead of cylinder crank		135°		225°	

The above gives us all the information to enable us to add to the main gear polygons lines covering the valve gear.

If, as I consider desirable in practice, the secondary polygons are drawn with the same length of side as the primary, and the reduction factor or connecting-rod ratio applied only to the resultants, the virtual secondary weights to be used in such case would be, for the engine in question (Conn. rod ratio = 4), four times as great as given above, the factor used being $1\ 196 \times 4 = 1\ 49$.

The moments follow directly from the weights, given the leverage of the valve center about the after crank.

LEVER DRIVEN AIR PUMPS.

3. Fig. 10, Plate 22, shows a very common type of appendage, namely, an air pump lever driven from the crosshead. Plate 23 shows diagrammatically the dimensions and weights.

The simplest method of procedure is to determine the combined center of gravity of the whole gear, which, being driven off the crosshead, shares in its motion. This is done in the table below:—

Wt.	Amount Lbs.	Leverage about Lever axis.	Moment about Lever axis.
Crosshead links.....	120	— 40″	— 4800
Levers.....	570	— 7″	— 3990
Pump links.....	195	20″	3900
Reciprocating pump gear...	1260	20″	25200
Sums.....	2145	9.47	20310

Now 2,145 pounds with a moment of 20,310 inch-pounds would be completely balanced by a weight at the crosshead end of the levers of 507.75 pounds. The effect of the air pumps as above, then, is to virtually reduce the reciprocating weight of the cylinder to which it is attached by 507.75 pounds. It will be observed that I neglected the obliquity of the links at the lever ends. This is because the effect of this obliquity upon the motion is infinitesimal.

BILGE PUMP ECCENTRIC DRIVEN.

4. The third and last type of auxiliary which I shall consider is that of a bilge pump eccentric driven.

Plate 24 shows diagrammatically such an appendage.

Stroke of engine is taken as 84 inches, which is seven times the pump stroke. Connecting-rod ratio for engine is 4 and for the pump is $6/30 = 5$. Center of pump is 10 feet forward of after crank center. In the case of this appendage we have a slight complication, owing to the fact that the bilge pump is below the shaft instead of above, as the cylinders are. A little consideration will make it obvious that primary polygons are not affected, but for secondary polygons the sides must be measured in the opposite direction to the crank direction obtained by doubling the actual angle, a result which is most conveniently reached by regarding the sides of the secondary polygons representing this appendage as negative. The methods applicable to this case are obvious, and the table below shows their application.

Items.	Actual Weight.	Regarded as Reciprocating.	Regarded as Revolving.
Eccentric sheave	275	—	275
Eccentric rod and strap	246	82	164
Reciprocating pump gear	460	460	—
Totals	—	542	439
Virtual primary weights—factor $\frac{1}{7}$	—	77	63
Virtual secondary weights—factor $\frac{1}{7} \times \frac{6}{30}$	—	−15.5	—
Virtual primary moments—factor $\frac{9}{7}$	—	774	627
Virtual secondary moments—factor $\frac{10}{11}$	—	−155	—

If, as I recommend, it is intended to draw secondary polygons with sides the same length as primary polygons, applying the connecting-rod ratio of the main gear, afterwards the factor for virtual secondary weights becomes $4/35$, and for virtual secondary moments $40/35$.

These make the virtual secondary weight for the reciprocating secondary force polygon 61.9, and the virtual secondary moment 619.

APPENDIX C.

Analysis and Discussion of Four-Crank Balance.

PRELIMINARY DEFINITIONS AND CONSIDERATIONS.

1. In taking up the question of the balancing possibilities, so to speak, of the four-crank engine, it seems desirable at first to cover the whole practicable field by determining crank angles and weights for primary balance throughout the range of possible cylinder spacings liable to occur in practice. In doing this I designate the cylinders by the numbers 1, 2, 3, and 4; No. 1 being aft, and No. 4 forward; then the reciprocating weights for the respective cylinders are denoted by w_1, w_2, w_3, w_4. The crank angles are measured in the direction of rotation from the direction of the forward crank. This is simply a matter of convenience. Any direction fixed with relation to the crank shaft could be used, but just as it is convenient to take moments about the after center, so it is convenient to measure angles from the forward crank. Another point to be noted in this connection is, that as long as the angles are measured in the same direction, they may be laid off either forward or backward; that is to say, all in the direction of rotation, or all opposite to the direction of rotation. This is a matter which may be of some importance when considering turning moments. The crank angles, then, referred to the direction of the forward or No. 4 crank, are denoted by a_1, a_2, a_3, and a_4, the latter being always = 0. As moments are taken about the after center, the distances or levers are measured from the after center. The distances, then, from the after center to the centers of the second, third, and fourth cylinders are denoted by l_2, l_3, and l_4. Since what we have to consider are ratios rather than absolute values, we may take any one reciprocating weight as always equal to unity and determine the ratios of the rest to it, and any one length as equal to unity and determine the ratios of the other lengths to it. It is evidently convenient to take the length of the engine from the after to the forward cylinder as unity—or l_4—equal to unity. It naturally follows, then, that we assume the forward reciprocating weight, or w_4, also equal to unity.

I invite attention now to Plate 32, which shows the force and moment polygons both primary and secondary for a four-cylinder engine, as described above. It is seen that the primary moment and primary force polygons are

both closed—that is to say, return to the origin, o. The secondary force and moment polygons are open—that is, there is a resultant. For convenience, the sides of the secondary polygons are the same length as the corresponding sides of the primary polygons, but it should be understood that the actual corresponding forces are, for the secondary, only $\frac{1}{n}$ those of the primary polygons where n is the connecting-rod ratio. The angles in every case are measured counter-clockwise from the direction OA. In the secondary polygons this may, and usually does, result in some angles being greater than 360 degrees. Thus, in the primary polygons, a_2 is about 255 degrees, so that $2a_2$ is about 510 degrees, or 360 plus 150.

DEDUCTION OF FORMULÆ FOR ANGLES AND WEIGHTS NOT ASSUMED AS KNOWN AND FOR RESULTANTS.

2. It is necessary next to deduce expressions or formulæ connecting the lengths of the sides and angles of the primary polygons, which are closed, and then to determine the resultants in the case of the open secondary polygons.

Consider, first, the primary moment polygon: This involves only the second, third, and fourth cylinders, since moments are taken about the first cylinder, and hence its moment is always zero. We assume that we know w_2, w_3, w_4, l_2, l_3, and l_4, and hence w_2l_2, w_3l_3, and w_4l_4. From well-known trigonometric formulæ we have from the primary moment polygon

$$\cos OAB = \frac{\overline{OA}^2 + \overline{AB}^2 - \overline{BO}^2}{2\ OA \times AB} = \frac{\overline{w_4l_4}^2 + \overline{w_3l_3}^2 - \overline{w_2l_2}^2}{2\ w_4l_4w_3l_3}.$$

Whence, as ω_4 and l_4 are each taken equal to unity,

$$\cos (180° - a_3) = \frac{1 + \overline{w_3l_3}^2 - \overline{w_2l_2}^2}{2\ w_3l_3}.$$

Similarly

$$BOA = a_2 - 180°.$$

$$\cos BOA = \cos (a_2 - 180°) = \frac{1 - \overline{w_3l_3}^2 + \overline{w_2l_2}^2}{2\ w_2l_2}.$$

The above determine a_2 and a_3 the only unknown quantities in connection with the second and third cylinders.

It is next necessary to determine the weight and crank angle for the

first cylinder. For this we use the primary force polygon. The formulæ are as follows :—

The polygon being closed, and each line being measured from the first extremity indicated to the second (*i. e.*, OA is a distance from O to A and AO is a distance from A to O), we have in the primary force polygon of Plate 32.

$$OA + AM + MN + NO = 0,$$

or

$$1 + w_3 \cos a_3 + w_2 \cos a_2 + w_1 \cos a_1 = 0.$$

Also,

$$MB + BP + CN = 0, \text{ or } w_3 \sin a_3 + w_2 \sin a_2 + w_1 \sin a_1 = 0.$$

Whence

$$\tan a_1 = \frac{w_3 \sin a_3 + w_2 \sin a_2}{1 + w_3 \cos a_3 + w_2 \cos a_2}$$

$$w_1 = \{(1 + w_3 \cos a_3 + w_2 \cos a_2)^2 + (w_3 \sin a_3 + w_2 \sin a_2)^2\}^{\frac{1}{2}}.$$

It is next necessary to determine the resultants in the secondary force and moment polygons. Referring to the open secondary force polygon, we have

$$OQ = OA + AM + PC + LD,$$
$$QD = MB + BP + CL.$$

The resultant OD is n times the unbalanced secondary force $= nf_2$ say. Denote the angle DOM by $2a_{f_2}$.

Then

$$OQ = nf_2 \cos 2a_{f_2} = 1 + w_3 \cos 2a_3 + w_2 \cos 2a_2 + w_1 \cos 2a_1,$$
$$QD = nf_2 \sin 2a_{f_2} = w_3 \sin 2a_3 + w_2 \sin 2a_2 + w_1 \sin 2a_1.$$

Whence

$$\tan 2a_{f_2} = \frac{w_3 \sin 2a_3 + w_2 \sin 2a_2 + w_1 \sin 2a_1}{1 + w_3 \cos 2a_3 + w_2 \cos 2a_2 + w_1 \cos 2a_1},$$

$$nf_2 = \{(1 + w_3 \cos 2a_3 + w_2 \cos 2a_2 + w_1 \cos 2a_1)^2 + (w_3 \sin 2a_3 + w_2 \sin 2a_2 + w_1 \sin 2a_1)^2\}^{\frac{1}{2}}.$$

Finally, for the determination of the resultant secondary moment,

referring to the open secondary moment polygon, we have, if nm_2 denote the secondary moment and $2a_{m_2}$ its angle

$$ON = OC \cos NOC = nm_2 \cos 2a_{m_2}$$
$$= OA + AM + PC = 1 + w_3 l_3 \cos 2a_3 + w_2 l_2 \cos 2a_2,$$
$$NC = OC \sin NOC = nm_2 \sin 2a_{m_2}$$
$$= MB + BP = w_3 l_3 \sin 2a_3 + w_2 l_2 \sin 2a_2.$$

Whence

$$\tan 2a_{m_2} = \frac{w_3 l_3 \sin 2a_3 + w_2 l_2 \sin 2a_2}{1 + w_3 l_3 \cos 2a_3 + w_2 l_2 \cos 2a_2},$$

$$nm_2 = \{(1 + w_3 l_3 \cos 2a_3 + w_2 l_2 \cos 2a_2)^2 + (w_3 l_3 \sin 2a_3 + w_2 l_2 \sin 2a_2)^2\}^{\frac{1}{2}}.$$

GRAPHIC REPRESENTATION OF RESULTS OF FORMULÆ AND CONCLUSIONS.

3. The above formulæ I applied to 256 cases, using four values of l_3, namely, .6, .65, .7 and .75; and four values of l_2, namely, .25, .3, .35 and .4. The weight of the fourth cylinder was always assumed equal to unity. The weights of the second and third cylinders were each assumed in succession .5, 1, 1.5 and 2. The final results are plotted on Plate 33, the "Diagram of Cylinder Spacings, Crank Angles, Weights, and Secondary Forces and Moments for Four-Crank Connecting Rod." In each little rectangle of this figure is shown an arrangement balanced as regards both primary and secondary forces. The lines marked 1, 2, 3 and 4 indicate, by their lengths, the reciprocating weights for the respective cylinders, and by their directions the directions of the corresponding cranks. The heavy full line shows in each case the amount of the unbalanced resultant secondary weight (it being assumed that the connecting rods are four cranks long), and the direction of the imaginary crank, with reciprocating weight developing secondary force alone, which corresponds to the resultant. The heavy dotted line similarly indicates the resultant secondary moment. Where a rectangle has no figure drawn, it indicates that primary balance was impossible with the assumed weights and cylinder spacing.

These diagrams give roughly an idea of the turning moments and the resultant secondary forces for all practicable four-cylinder engines balanced primarily. An inspection of them indicates generally that where the crank angles are such as to give an evidently undesirable turning moment, the resultant secondary force is usually large, and moreover this condition is apt to be accompanied by differences in the reciprocating weights so large as to be impracticable. The resultant secondary force varies a great deal

more both in direction and magnitude than the resultant secondary moment. It should be understood, in considering the resultant secondary moment, that the length of the line 4 in every case represents the moment of reciprocating weight 4 about the after crank.

A final fact, which is evident from the diagram, is that the best results as regards turning moment and residuary secondary force and moment appear to be developed when the second and third weights are both somewhere near 1.5. When this is the case, the first weight is generally not far from unity, or the value of the fourth weight, and in general the distribution of crank angles and weights is good. It is evident from this diagram that detailed investigation then may be restricted to combinations in which the angle of the first crank is from about 40 degrees to about 60 degrees; the angle of the second is something less than 270 degrees, and the angle of the third some 20 to 40 degrees less than 180 degrees. It is evident, moreover, from the diagram, that it will be practically impossible to eliminate altogether resultant secondary moments, while it appears probable that resultant secondary forces can be eliminated. The problem is then to determine the combinations where primary forces and moments are all eliminated, and then determine the corresponding resultant secondary moments. I have found that this can be done without a prohibitive amount of labor by a new graphic process.

DETERMINATION OF DIAGRAMS FOR FOUR-CRANK BALANCE, WHICH IS COMPLETE EXCEPT AS TO SECONDARY MOMENTS.

4. Referring to Fig. 26, Plate 34, suppose that OBC is a closed moment polygon, and $OBDF$ the three corresponding sides of a primary force polygon, then we have

From the moment polygon:—

$$OB = w_4 l_4$$
$$BC = w_3 l_3$$
$$CO = w_2 l_2$$
$$OB = BC \frac{w_4 l_4}{w_3 l_3}.$$

From the force polygon:—

$$OB = w_4$$
$$BD = w_3$$
$$DF = w_2$$
$$OB = BD \frac{w_4}{w_3}.$$

Whence

$$BD \frac{w_4}{w_3} = BC \frac{w_4 l_4}{w_3 l_3}$$
$$BD = BC \frac{l_4}{l_3}$$
$$CD = BD - BC = BC \frac{l_4 - l_3}{l_3}$$

Now by similar triangles

$$\frac{OE}{OB} = \frac{CD}{CB} = \frac{l_4 - l_3}{l_3}.$$

Whence for given relative values of l_4 and l_3 and a constant value of OB or w_4, the line DF of the force polygon representing w_2 passes always through a fixed point E on BO extended. Furthermore, DF is divided at the point E in a ratio depending on l_2 and l_3.

For

$$DF = OC\,\frac{l_4}{l_2}\,.\;DE = OC\,\frac{l_4}{l_3}.$$

Whence, finally,

$$\frac{DE}{EF} = \frac{l_2}{l_3 - l_2}.$$

Evidently, then, if in drawing a primary force polygon, given ω_4, l_4, l_3 and l_2, we determine the point E from the formula

$$\frac{OE}{OB} = \frac{l_4 - l_3}{l_3},$$

and then draw a line DF at any inclination through E, determine D and F from the formula $\frac{DE}{EF} = \frac{l_2}{l_3 - l_2}$, then join FO, DB, we have a closed primary force polygon for which the primary moment polygon is also closed. The primary moment polygon is given at once by drawing OC through O parallel to DF.

Since we can draw any number of lines, such as DF, we can determine any number of weights and angles giving primary balance for given lengths of engine, given forward reciprocating weight, and given position of the second and third cylinders. This being the case, we can impose another condition. For instance, we can so determine the angles and corresponding weights as to get a desirable turning force, *i. e.*, the torque as uniform as possible during a revolution. Or we can determine weights and angles which reduce the secondary force to zero or *close* the secondary force polygon. This is readily done by trial and error. Assume a value and position of DF and draw the corresponding secondary force polygon. Guided by this, assume a second value and position of DF, for which the secondary force polygon is more nearly closed, and so on. I found in practice that after about three successive values of DF I could determine the value giving a closed secondary force polygon. It was in this way that I obtained data

from which to plot the Diagrams A to F, giving values of a_1, a_2, a_3, w_1, w_2 and w_3. These diagrams are practically correct. After they were plotted and traced, I took off values of a_1, a_2, etc., for 35 values of l_2 and l_3 and *calculated* the resultants of primary and secondary force and moment polygons. The primary and secondary force and primary moment polygons were practically closed in every case. The results as regards the secondary moments are given in Diagram G. Fig. 27, Plate 34, shows a typical case. The primary moment polygon is closed, as are the primary and secondary force polygons. The secondary moment polygon has a resultant OC, the length of which on the scale used for the primary moment polygon divided by n, the connecting rod ratio, denotes the actual unbalanced moment M_2. Note that in the Diagram G, giving residual secondary moments, the scale values represent nM_2, so that this can be used for connecting rods of any length ratio.

MODIFICATIONS NECESSARY BECAUSE OF VALVE GEARS, ETC.

5. The six weight and angle diagrams for the determination of a_1, a_2, a_3, w_1, w_2, and w_3, can, of course, be used for the main reciprocating weights only. When considering the valve gears and other secondary weights, additional work is necessary, which may be briefly outlined as follows:—

From the given cylinder locations determine the weight ratios. From the weight ratios determine the actual weights neglecting valve gears. The angle diagrams A, C, and E give the first approximation to each crank angle, and hence to each eccentric angle since the angle of advance is known for each eccentric. From these draw the primary and secondary force and moment polygons of the valve gear. These will, in general, have resultants. Now determine new weights and angles for the main gears, such that they are not in balance but *balance* the valve gears. This is a second approximation. Continue the process until the results are practically exact. The process is comparatively short and simple, and gives substantially exact results.

SOME GENERAL CONCLUSIONS.

6. An inspection of the weight and angle diagrams shows that the weights for the middle or heavy cranks are apt to be nearly the same, while those for the end cranks differ materially and are lighter than either of the others. That is, if applied to quadruple engines, the H. P. and first M. P. should be at the ends of the engine. The diagrams are drawn primarily

for No. 1 cylinder aft, No. 4 forward, and the crank angles measured in the direction of rotation of the engine. It is quite permissible, however,

1. To lay off all crank angles opposite to the direction of rotation. This follows from the fact that a balanced engine is balanced for rotation either ahead or astern. To gain a better curve of torque it may be desirable to set off crank angles in one direction rather than the other.

2. To turn the engine bodily end for end. This will often be desirable in order to bring the lightest cylinder—the H. P.—at the forward end of the engine.

DISCUSSION.

THE CHAIRMAN:—We will not take up the discussion of this paper at the present time as the hour for adjournment for lunch has arrived. The first paper for the afternoon will be the paper which received the second prize, and, after it has been presented, discussion will be in order on both papers. When we adjourn it will be until 2 o'clock.

The meeting then adjourned until 2 P. M.

SESSION OF THURSDAY AFTERNOON.

The meeting was called to order at 2.30 P. M., Mr. Stevenson Taylor in the chair.

THE CHAIRMAN:—We will now listen to the reading of the fifth paper on the programme—the second prize competition paper—on the Balancing of Marine Engines, which is sent to us by Rear-Admiral Melville. I understand Mr. McFarland is prepared to present this paper. Please give him your attention.

MR. WALTER M. MCFARLAND, *Member of Council*:—Gentlemen, I know, of course, it must be a matter of great regret to all of you that the author of the paper is not here himself with his striking and delightful personality to present it to you, instead of such a poor substitute. But I was for so many years associated with him that he counts me as one of his boys even now, although it is some three years since I actually left his paternal care, and I think I am fully authorized to say in his behalf—I know he would have said it if he could have been here—how much he would have enjoyed hearing the paper that was read just before luncheon, which certainly is one of the most admirable papers that has ever been presented on the subject of

marine engine balancing. I said to Mr. Taylor, when I congratulated him personally just before lunch, that I thought he had been exceedingly modest in that in his own paper he made no reference whatever to the fact that, in a sense, he is, as you may say, almost the author of this system of balancing by a 4-crank engine. Some eleven years ago Mr. Taylor read a paper which is published in the journal of another Society, where he went into this subject very fully, and gave a very careful and complete analysis of circumstances attending engine balancing, and I may say, in fact, that this paper was so complete that when the Yarrow-Schlick-Tweedy people came over here to get a patent, they ran against the fact that Mr. Taylor, while he had not taken out a patent, had really covered the whole subject, and covered it very fully. In Admiral Melville's paper, of which I saw the carbon copy before he sent it on, I remember tha t he says on page 97 : —" Those wishing to follow the matter further may consult the article on 'The Causes of Vibration of Screw Steamers,' by Assistant Naval Constructor D. W. Taylor, U. S. N., in the Journal of the American Society of Naval Engineers, Vol. III, p. 20. Constructor Taylor was the first to propose a 4-crank engine with unequal crank angles in order that the tendency to vibration might be 'practically annulled.'" As I had occasion to say here once before, when reading a paper prepared by someone else, it is considered a prerequisite of membership in this Society that one should be able to read and write, and therefore, as the paper is in print, I will not undertake to read the whole thing, but will try to give you a brief digest of it.

Mr. McFarland then gave an abstract of Admiral Melville's paper.

THE THEORETICAL AND PRACTICAL METHODS OF BALANCING MARINE ENGINES.

By George W. Melville, Rear-Admiral and Engineer-in-Chief, U. S. Navy, Vice-President.

[Read at the ninth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 14 and 15, 1901.]

The balancing of engines has been familiar to the engineering profession for a great many years, particularly in locomotive engines and very high speed engines on shore. Within the past fifteen years the subject has attracted much more attention among marine engineers on account of the effect of lack of balance in the engine in causing disagreeable vibrations in the hull. The matter has been brought before various engineering societies by some of the most accomplished leaders in the profession, and several systems have been actually applied in practice, while still others have been advanced with strong arguments to show their capacity to solve the problem of engine balancing and thereby that of hull vibration.

The system which has thus far been applied to the greatest extent in practice is the Yarrow-Schlick-Tweedy system of four cylinders with unequal crank angles and other arrangements for preventing vibration. While this system has been partly successful in accomplishing its aim, competent observers on the ships to which it is fitted have found that it is not a complete solution of the problem, and, indeed, in some cases the vibration has been said to be almost as bad as with unbalanced engines.

Mr. John H. Macalpine, who has published a number of articles on this subject, is a personal friend of the writer, and has favored him with very complete information with respect to a system devised by Mr. Macalpine, and which is a complete solution of the problem of balancing a marine engine.

The members of this Society are, of course, well aware that our Navy has not adopted the Y. S. T. system of engine balancing, and as the writer is in a position to know the reasons which have actuated the Engineer-in-Chief of the Navy in his decision, he believes it will be of interest to present to the Society a discussion of the various methods which have been proposed or actually tried for balancing marine engines, including the Y. S. T. system, giving their merits and demerits, and going at considerable length into

the Macalpine system of balancing, which the writer believes to be the best solution of the problem thus far presented. Obviously, in a discussion which aims to give a history of the subject, prominence will be given not only to systems which are correct, but to those which are only partially so.

Unbalanced engines do not at all times vibrate a ship, but only at particular speeds of revolution. This is a well-known fact which can be readily observed by everyone who has been on board ship to any great extent—one which has been clearly brought out by many writers on our present subject. For instance, Mr. Yarrow says, in his paper read in 1892 before the Institute of Naval Architects:—"Some years since, we had a boat in which severe vibration occurred at 200, 400, 600, and 800 revolutions per minute, but there was none at the intermediate speeds of 300, 500, and 700."

Thus there are the proverbial two sides to this question. The engine must supply impulses to the ship at or near the rate at which the ship itself would naturally vibrate. In other words, there must be synchronism between the speed of revolution of the engine and the free vibration of the ship. We may recall the old illustration of soldiers marching over a suspension bridge, where, if the step corresponds with the rate of vertical swing of the elastic structure, there is danger of breaking down the bridge by producing a large vibration. If the rate of march is increased or diminished, the vibration of the bridge will diminish, but the usual practice is to make the soldiers "break step," so as to avoid any chance of the dangerous timing of the impulses.

If, in the ship, we could always be sure of avoiding these troublesome speeds of revolution, there would be little need for accurately balanced engines. But it would be far too serious a restriction on the use of the engines, and it is impossible to predict with any sufficient degree of accuracy what the period of vibration of the ship will be. Besides, Mr. Yarrow's experiments bring out what is otherwise very well known, that the ship is capable of responding to very different speeds of impulse—or, in other words, it may vibrate in many different ways, each having a period peculiar to itself.

It is now fully recognized that a reciprocating engine of the usual kind does not supply merely one set of impulses at the same rate as it revolves, but a number of different sets of sensible magnitude and various periods. This makes the problem of preventing vibration by the adoption of a particular speed of engine a greatly complicated one.

Thus the whole problem is complicated mathematically, whether we consider the effect produced on the ship or the forces we have to deal with in the engines.

If we can produce an engine which, when placed in the comparatively delicate elastic structure of the ship, will have all its inertia forces balanced, so that they do not strain the ship, or if only those are transmitted to the hull which will not nearly synchronize in period with vibrations of a kind they could excite, consideration of the effect on the ship practically ceases to be part of the problem.

To produce such a balanced engine we must adequately meet the comparatively complex dynamical conditions in the engine, and the solution must be not merely a theoretical one, but it must result in an engine which will not fall behind our best marine engines of to-day in steam consumption, and must be equal to them in first cost, upkeep, and practicability generally.

If it was merely a diagram or design on paper that was wanted, it would be easy to produce a great variety of "theoretically" balanced engines. If, for instance, we were allowed to "suppose" an infinite connecting rod, or to "suppose" that a slotted crosshead would work as well as a connecting rod, all we require is to make an engine with three cranks in one plane, the center one standing up when the outer ones were pointing down.

Counterbalance the cranks and properly adjust the reciprocating weights, and the balance is perfect. The engine would not start from its dead points, but that would be found to be only one of its minor defects.

Such sentences are, no doubt, trivial. Unfortunately, they are not more so than many of the proposals that have seriously been put forward in this connection, all of which we may dismiss as impracticable without any further notice. We are not at liberty to make suppositions. We have our limits of weight and space; the dynamical problem is definite, and the inexorable laws of nature will be our judge whether we have clearly comprehended the conditions or sufficiently submitted to them in carrying our theory into practice. It is a theoretical *and* practical problem, and both sides must be fully attended to. Or, in the quotation very familiar in this country, "It is a condition and not a theory which confronts us."

Much that deserves notice has proceeded on the assumption that the connecting rod might be treated as of infinite length, but, unfortunately, this leads to the omission of forces of considerable magnitude. If of infinite length, the inertia of the piston and other reciprocating weights would be of the same intensity at each end of the stroke, say at top and bottom stroke, in a vertical marine engine. Designate this force by F. Now let us see the effect of a connecting rod of finite length. It has long been known that if r is the crank radius and l the length of the connecting rod, the actual forces are:—

At top stroke, $F\left[1 + \frac{r}{l}\right] = F \times \frac{5}{4}$ where $l = 4r$.

At bottom stroke, $F\left[1 - \frac{r}{l}\right] = F \times \frac{3}{4}$ where $l = 4r$.

So that, for the length of connecting rod very commonly employed, the forces at top and bottom stroke, instead of being equal, as supposed, actually bear the ratio of 5 to 3; the force at top is nearly double the other. Any system which does not take account of this difference cannot be considered perfect. It may be looked on as a first approximation, but an approximation far from satisfactory.

We have this defect in the Wells' engine, a type of engine which has been patented by many inventors under the impression it was balanced. In it, two cylinders are arranged, one above the other, in line. The upper and lower pistons are actuated by exactly opposite cranks, and the *intention* is that the inertia of the lower piston and attached moving parts at, say, bottom stroke, will balance the inertia from the upper piston and its attached parts at top stroke. The intention is far from being fulfilled, as we have just seen, and it was, doubtless, for this reason that Admiral Melville, a number of years ago, declined to approve it. The engine has some good points, but I think that this defect is what has hindered its progress.

When the Y. S. T. system was proposed, it was not suggested that it in any degree took account of the effect of the finite length of the connecting rod. That it does not do so will be shown fully in the sequel. This is the reason for the non-adoption of this system by our Navy, and I believe that actual experience has justified this course.

I will now develop the theory of the subject so far as to explain sufficiently all of value that has been done, and will give the reasons for my belief that the Macalpine system of balancing is the best.

The fundamental problem may be said to be the counterbalancing of a simple revolving piece, and with this I will commence.

Suppose a revolving shaft to have a projecting arm or crank. The centrifugal force of this part will tend to deflect the shaft in the direction of the arm. This centrifugal force is readily calculated. If M_1 is the mass of the arm in pounds, and R_1 the radius of its center of gravity from the shaft center, in feet, also, if ω is the angular velocity of the shaft in radians per second, and g the intensity of gravity, the whole centrifugal force, as is well known, is

$$F_1 = \frac{M_1 \omega^2 R_1}{g}, \qquad (1)$$

F_1 being also in pounds.

Now, to counteract this centrifugal force, all we need do is to place a counterbalancing arm on the opposite side of the shaft, the line through the centers of gravity of the original and counterbalancing arms being at right angles to the axis of the shaft. If, then, M_2 and R_2 are the mass and radius of the center of gravity for the counterbalance, we have the centrifugal force for it

$$F_2 = \frac{M_2\omega^2 R_2}{g}, \tag{2}$$

and to get an exact balance we make

$$M_2R_2 = M_1R_1, \tag{3}$$

We can thus exactly balance the cranks of any engine and the parts revolving with them, such as the lower part of the connecting rod.

This would leave the vertical inertia forces from the upper part of the connecting rod and the parts above it entirely unbalanced, and these would cause vertical vibrations of the ship. Now, it is the vertical vibrations that are principally troublesome. Even when the cranks are uncounterbalanced, the horizontal forces rarely have produced troublesome horizontal vibrations, the principal reason being that the ship is much stiffer when bent in a horizontal than in a vertical plane. Probably, also, the waves produced carry away more readily the energy of horizontal than of vertical vibrations.

The vertical downward force from the counterbalance is

$$F_2 \cos\theta = \frac{M_2\omega^2 R_2}{g} \cos\theta, \tag{4}$$

where θ is the angle by which the crank has passed top center. This is obtained at once by resolving vertically the whole radial force from the counterbalance.

It is a force due to a simple harmonic motion of the same period as the engine revolution. If the connecting rod were of infinite length, the piston and other reciprocating parts would also have a simple harmonic motion of this period, and the intensity of the upward force from them would be

$$F = \frac{M\omega^2 R}{g} \cos\theta, \tag{5}$$

where M is their total reciprocating mass and R the crank radius.

As the forces in equations (4) and (5) follow the same law of variation, the whole vertical inertia force could be counterbalanced if the connecting rod could be made of infinite length. Now, if we examine the differences of the actual motion of the piston, and that it would have were the connecting

rod infinite, we would find that these differences followed quite a different law of variation from that of the above equations. Hence the simple harmonic motion, got by supposing an infinite connecting rod, is only an element of the actual motion with the connecting rod finite, and it is the only part of it which can be balanced by a counterbalance.

Thus we may make the counterbalance such that

$$M_2R_2 = M_1R_1 + MR. \qquad (6)$$

We will then have that part of the whole vertical force, which has a simple harmonic motion of the same period as the engine, counterbalanced; but the horizontal forces will be overbalanced.

As MR is not usually greatly different from M_1R_1, the correct horizontal balance will have been almost doubled, or, otherwise stated, the unbalanced horizontal forces will now be about as great as when no counterbalances were fitted, but in the opposite direction. Since, as noted above, the horizontal unbalanced forces do not usually cause trouble, there is no very great objection from this point of view to a counterbalance as large as given by equation (6).

Practice varies between the limits given by equations (3) and (6). One practical objection to the larger value, besides the very considerable addition to the weight of the engine, is that, if steaming at reduced initial pressure, the crank shaft may never be lifted off the lower brass of the main journals, thus causing imperfect lubrication, heating, and rapid wear.

Instead of counterbalances on the cranks, revolving arms, placed usually forward and aft of the engine, have been fitted. In most cases, the total weight of counterbalance will be considerably smaller than if each crank is separately dealt with. Their exact position being determined to suit the design, their mass, radius, and the angles at which they stand to the cranks can readily be determined. This can be done either entirely by calculation or by a semi-graphical process. It will be sufficient to indicate the former method here.

Let the cranks be designated by the numbers 1, 2, 3, 4, etc., in order from the forward end. m_1, m_2, m_3, m_4, etc., are the corresponding moving masses. That is, the mass of the crank is to be reduced to crank radius, so that instead of M_2 of equation (1) we take $\frac{M_1R_1}{R}$. To this is to be added the whole of the connecting rod and parts moving with the crosshead to make up m.

a_2, a_3, a_4, etc., are the angles by which the cranks 2, 3, and 4, etc.,

respectively, lead crank 1, and θ is now the angle by which crank 1 has passed top center.

The positions of the counterbalances, chosen to suit the design, will be known, as will also the values of the m's and a's and the fore-and-aft positions of the cranks. We may then express the moment of all the forces (including valve gears or any other attached parts, which will be reduced to crank radius and treated exactly as the cranks) about the forward counterbalance. For a balance, this moment must always be zero. Take d_1, d_2, d_3, etc., as the distance the respective crank centers are aft of the forward counterbalance; and let n, β, e, and r be the mass, angular advance, distance, and radius of center of gravity of the aft counterbalance. The moment then is

$$\frac{\omega^2}{g}\left[R\{m_1d_1\cos\theta + m_2d_2\cos(\theta+a_2) + m_3d_3\cos(\theta+a_3) + \text{etc.}\} + ner\cos(\theta+\beta)\right] = 0. \quad (7)$$

For the balance required equation (7) must always be zero; and, as is well known, this will hold if it holds for two values of θ not differing by two right angles. Take $\theta = 0$ and $\theta = \frac{\pi}{2}$. Then

$$\theta = 0 \;.\; nr\cos\beta = -\frac{R}{e}(m_1d_1 + m_2d_2\cos a_2 + m_3d_3\cos a_3 + \text{etc.}) = A, \quad (8)$$

$$\theta = \frac{\pi}{2} \;.\; nr\sin\beta = -\frac{R}{e}(+ m_2d_2\sin a_2 + m_3d_3\sin a_3 + \text{etc.}) = B. \quad (9)$$

As all the quantities entering into A and B are known, we can at once calculate nr and β thus,

$$nr = (A^2 + B^2)^{\frac{1}{2}},\ \tan\beta = \frac{B}{A}. \quad (10)$$

In the same way we may calculate the values for the forward counterbalance. The engine will now be balanced exactly for those vertical forces which have a period equal to the time of revolution of the engine.

Although Herr Schlick had read a most interesting paper in 1884, before the Institute of Naval Architects, on the "Vibrations of Steam Vessels," the real impetus to the present development of the allied subjects of vibrations of steamships and engine balancing was given by a justly famous paper read by Mr. Yarrow before the same Society in 1892, on "Balancing Marine Engines and Vibrations of Vessels." There he shows how, by a calculation und amentally the same as that in equations (7) to (10) we may determine

the mass and stroke of "bob-weights," placed forward and aft of any engine; and the angular position, relative to the cranks, of the eccentrics driving them, so that the same vertical balance, as given above for the revolving counterbalances, may be attained. Obviously, nr and β are the same whether we adopt revolving counterbalances or bob-weights driven by eccentrics. The only advantage of the latter is that the excessive horizontal forces are avoided. In fact, if we counterbalance the revolving parts by revolving counterbalances, and the reciprocating parts, only, by bob-weights, an exact balance both of horizontal and vertical forces (connecting rod infinite) may be attained.

As the throw of the eccentrics is necessarily very limited, the bob-weights become comparatively heavy; and, I believe, Mr. Yarrow very soon substituted the lighter revolving masses which could have a greater travel given to them, thus returning to the old counterbalancing.

Instead of driving idle bob-weights by eccentrics, to produce balancing forces by their inertia, we may evidently transform the eccentric into a crank and make the bob-weight into the piston, etc., working in a cylinder. Its first function will not be interfered with by this change. When this is realized in a four-crank engine, we have the Y. S. T. system. I will only give the equations referring to this system. Using the same notation as before, we have for the total upward force,

$$\frac{\omega^2 R}{g}\{m_1 \cos\theta + m_2 \cos(\theta + a_2) + m_3 \cos(\theta + a_3) + m_4 \cos(\theta + a_4)\}. \qquad (11)$$

Similar terms being added to include the mass of the valve gears, etc., reduced the crank radius.

Taking the moments about the forward crank and measuring d_2, d_3, d_4 from the center of this crank, we get the equation for moments,

$$\frac{\omega^2 R}{g}\{d_2 m_2 \cos(\theta + a_2) + d_3 m_3 \cos(\theta + a_3) + d_4 m_4 \cos(\theta + a_4)\}. \qquad (12)$$

To obtain a balance we must equate these to zero at two positions of the revolution, say, at $\theta = 0$ and $\theta = \frac{\pi}{2}$. We then get the four equations.

$$\left.\begin{aligned}
&\text{From (11), } \theta = 0 . && m_1 + m_2 \cos a_2 + m_3 \cos a_3 + m_4 \cos a_4 = 0 \\
&\text{From (11), } \theta = \tfrac{\pi}{2} . && m_2 \sin a_2 + m_3 \sin a_3 + m_4 \sin a_4 = 0 \\
&\text{From (12), } \theta = 0 . && d_2 m_2 \cos a_2 + d_3 m_3 \cos a_3 + d_4 m_4 \cos a_4 = 0 \\
&\text{From (12), } \theta = \tfrac{\pi}{2} . && d_2 m_2 \sin a_2 + d_3 m_3 \sin a_3 + d_4 m_4 \sin a_4 = 0
\end{aligned}\right\} \qquad (13)$$

We may choose three of the quantities, say, m_1, m_4, and a_4, and by solving the equations find corresponding values of m_2, m_3, a_2, a_3; d_3 and d_4 will be practically determined by the design.

I merely give equations (13) to show clearly the scope of the Y. S. T. system and its limitations; and also for a subsequent remark on the effect of short period forces. I do not give methods of solution, as I am not advocating the scheme. Those wishing to follow the matter further may consult the article on "The Causes of Vibration of Screw Steamers," by Assistant Naval Constructor D. W. Taylor, U. S. N., in the Journal of the American Society of Naval Engineers, Vol. III, p. 20. Constructor Taylor was the first to propose a four-crank engine with unequal crank angles, in order that the tendency to vibration might be "practically annulled." Also, see Herr Schlick's paper, "Further investigations of the Vibrations of Steamers," Institute Naval Architects, 1894. Admirable methods of semi-graphical solution, which can be applied to this system, including valve gears, are given by Professor Dalby in a paper, "The Balancing of Engines, with Special Reference to Marine Work," Institute Naval Architects, 1899, and in one by Mr. J. Macfarlane Gray, "The Geometry of Engine Balancing," read before the same Society this spring (1901).

If the cranks are so far counterbalanced as to make the masses of the revolving parts reduced to the crank radius—that is, the webs, pin, and part of the connecting rod—proportional to m_1, m_2, m_3, m_4, evidently these revolving parts are balanced among themselves at all parts of the revolution. Thus the horizontal as well as the long period vertical forces are balanced. I believe this horizontal balance is not usually carried out in practice, though this is the *only* advantage that can possibly be claimed by the Y. S. T. system over the older counterbalancing by revolving weights at the end of the engine. As already shown, this latter leaves unbalanced horizontal forces, but such as rarely give trouble.

The revolving counterbalances, on the other hand, have the advantage of great simplicity, and the avoidance, early in the design, of the troublesome determination of crank angles and adjustment of moving masses, with the attendant difficulty of making different parts of the crank shaft replaceable by one spare section. Also, we are not tied down to a four-crank engine.

That the Y. S. T. system does not give a perfect balance, and that the difficulties above mentioned are not imaginary, are brought out more strongly by the fact that the system has only been used to a limited extent, and where it was important to claim that at least an effort had been made

to solve the problem. We will incidentally examine it a little further in treating the effects of the forces which pass through two complete cycles in one revolution of the engine. These are the forces which principally account for the inertia at top stroke being so much greater than at bottom stroke. These short-period forces are not produced at all by revolving counterbalances, which is much in their favor.

It has long been known that the inertia from the parts reciprocating with the piston is very approximately given by

$$F = \frac{M\omega^2 R}{g}\left\{\cos\theta + \frac{r}{l}\cos 2\theta\right\}.$$

All that has thus far been discussed neglects the very important second term of the equation just given.

I may at once give the exact formula, taken from Mr. J. H. Macalpine's paper, read before the Institute of Naval Architects this summer, entitled "A Solution of the Vibration Problem." And I may add that, as this paper gives such an excellent exposition of the Macalpine System, I shall quote from it very liberally in what follows. The numerical coefficients are for a connecting rod four cranks in length.

$$F = \frac{M\omega^2 R}{g}(\cos\theta + .2540\cos 2\theta - .0041\cos 4\theta + .00007\cos 6\theta + \text{etc.}) \quad (14)$$

We must now deal with the short-period forces expressed by the terms after $\cos\theta$ in the infinite series of equation (14). I think their existence and nature had not been recognized when Mr. Yarrow read his paper in 1892. As establishing their importance and, at least, usual origin in the engine, I will make a quotation from Mr. Macalpine's paper referred to above. He says:

"The second factor of equation (1)* is an infinite series, but the only term considered by Mr. Yarrow is the first, viz., $\cos\theta$. On remarking to Mr. Yarrow [in 1892] that the complete series should give rise to other and shorter period vibrations than those occurring at the same speed as the engine, he at once said that explained the peculiar form of wave produced by the vibrations of the boat, these waves having sharp crests and well-rounded hollows.

"Thus, from the first, we have strong evidence of the marked presence

* Equation (14) of the present paper.

of these short-period vibrations arising from the engine—for in many of Mr. Yarrow's experiments the propeller was disconnected.

"In the same year, 1892, Mr. H. C. Flood and I made observations on the Circassia, and found not only vibrations of the same period as that of the revolution of the engine, but also, very prominently, those of one-half and one-fourth of that period. These we called the 1st, 2d, and 4th period vibrations. There were also marked indications of vibrations of the 6th and 8th period, but no trace of any odd period. This is precisely what we would expect from equation (1), which contains, besides cos θ, only the cosines of even multiples of θ. In several other respects, detailed in our paper, the observations confirmed what theory would lead us to expect; and I think they leave no doubt that these vibrations originate in the engine.

"Scarcely has there been a vibration diagram published which has not borne the clearest evidence of those of short period; and no one who has followed the development of this whole subject could fail to be struck by the growing recognition of their importance. Indeed, in two cases which came under my own observation, and in one which was observed by a friend, the second period vibration was much more prominent than that of the first period.

"I maintain that the short-period forces *must* be balanced in any completely satisfactory solution of the vibration problem, and this *cannot* be done with the ordinary type of direct-connected marine engine."

Perhaps no stronger proof could be given of the importance of short-period vibrations than the fact that Herr Schlick, who has made so many excellent observations of vibrations of ships, read an elaborate paper in 1900 before the Institute of Naval Architects, to prove that the Y. S. T. engine could be practically balanced for second period. With this claim, however, I cannot agree, and I believe it has not been substantiated. The reasons for this I will give shortly.

The importance of these high-period vibrations being now thoroughly recognized, I will give a simple method of judging quickly the degree of balance or unbalance an engine has with regard to any particular period, and use it in two cases. This method has been used both by Mr. Macalpine and by Professor Dalby.

The second-period forces and couples are given by expressions similar to (11) and (12). That for the forces is

$$\frac{.2540\ \omega^2 R}{g} \times$$

$$\{m'_1 \cos 2\theta + m'_2 \cos 2(\theta + a_2) + m'_3 \cos 2(\theta + a_3) + m'_4 \cos 2(\theta + a_4)\}. \quad (15)$$

I have replaced m by m', as only the parts moving with the crosshead, and part of the connecting rod give rise to higher period forces. Beyond this difference and the introduction of the factor .2540, this expression only differs from (11) by all angles being doubled. If, then, starting from any particular engine A we keep the forward crank at top center and double all the angles by which the other cranks and eccentrics stand to the forward crank, we get a new engine B. Neglecting differences of m and m' in expressions (11) and (15), engine B will have the same degree of balance or unbalance for first period forces and couples as engine A has for second period. Similarly, by quadrupling the crank angles we get an arrangement by which the balance or unbalance of fourth period may be easily judged. And so on for all periods.

1st. Let us apply this to the crank angles of the Deutschland, which is balanced on the Y. S. T. plan.

Deutschland's crank angles.

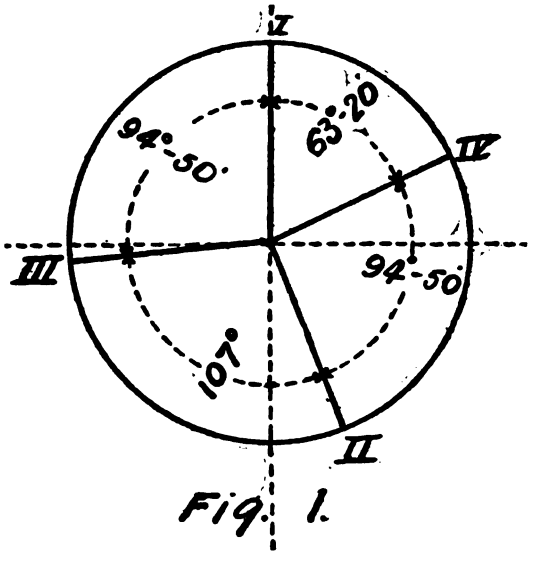


Fig. 1.

Deutschland's crank angles doubled.

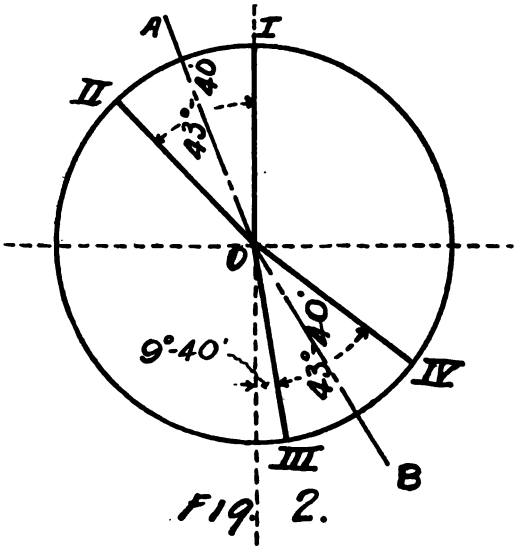


Fig. 2.

Fig. 1 gives the crank angles taken from Herr Schlick's paper, "On Some Experiments Made on Board the Atlantic Liner Deutschland during her Trial Trip in June, 1900," read at the Spring Meeting, Institute Naval Architects, 1901.

Fig. 2 gives the crank angles doubled. This brings cranks I and II both near top center when cranks III and IV are both near bottom center. The second period couple is, therefore, very large. If cranks I and II had lain in the direction OA, and cranks III and IV in the direction OB (these

directions bisecting the angles between the pairs of cranks), the second period couple would have been within half a per cent. of the greatest possible.*

In Fig. 2 the angles of the cranks give a couple, less than this in about the proportion $\cos \frac{1}{2}(43° - 40') = .928$. (The exact value of this couple could only be found after knowing the masses attached to cranks I, II, III, and IV, but it would differ little from the above.†) Thus the Y. S. T. system gives a second period couple not far from the worst possible; and this will be found always to be the case in this system. Herr Schlick's, 1900, Institute Naval Architects' paper, was written to prove that, besides the first period balance, the *force* of second period could be balanced also. But as this *force* is always of small amount, surely its exact balance is of little importance when so large a *couple* is left unbalanced. Noticing these couples at the end of his paper, Herr Schlick says:—

"It is evident from the foregoing that there always remains a small couple of forces, and it might be feared that these would cause vibrations. This is, however, in most cases, practically impossible. As in one revolution of the crank shaft two maxima and two minima of equal magnitude arise, therefore the vibrations of the longitudinal axis of the steamer could only be produced when the number of revolutions of the engine is only half that of the vibrations in the longitudinal axis, viz., when the engine is working dead slow. But with half the number of revolutions, the couples which arise are only one-fourth of the values given above."

Surely this is a remarkable statement. If true, would it not apply equally to the second period *force?* In which case the reason for the whole paper disappears; especially as, in all cases on the Y. S. T. system, the second period force is never more than a very small fraction of the greatest combined force of this period that the four cranks could exert, since the second period forces from the various cranks partly oppose one another; while, on the other hand, as just shown, the second period couple is near the greatest possible value. It may further be asked if any competent observer has ever seen a vibration diagram that did not give clear evidence

* Cos $\frac{1}{2}AOB = -\cos \frac{1}{2}(9°-40') = -.996$. If AOB had been in a straight line, the couple would, obviously, have been the largest possible.

† Figs. A to D, Plate 46, were drawn, and the text referring to them was written after the rest of the paper was sent in. The masses attached to the cranks are given there and the exact reduction of couple, given approximately above, could at once be deduced. But the figure given is sufficient for the moment.

of the simultaneous presence of more than one period of vibration, thus showing that the ship was vibrating in more than one mode at the same time; responding, on account of approximate synchronism, to unbalanced forces or couples of various periods? If so, would he say that the engine must run at half speed before there will be any response to the unbalanced second period couples?

The above statement, indeed the reason for the whole of his paper, is more difficult to understand in view of a statement in his 1901 paper, that—

"Most of the vessels supplied with my system of balanced engines have been run for trial with the propellers uncoupled to examine the effect of balancing. On all these trials vibrations have never been noticed, even when the engine reached the critical number of revolutions; whereas engines not balanced always show excessive vibrations when running with disconnected propellers, as soon as the critical number of revolutions is reached."

If this quotation was strictly accurate, it would be difficult to understand the object of writing a long paper to discuss the exact arrangement of cranks by which a small second period force may be annulled. The fact is, if our discussion thus far is correct, that engines on the Y. S. T. plan are not perfectly balanced, and we find it difficult to believe that they should not noticeably vibrate the ship when the propellers are disconnected, as it is alleged other unbalanced engines do, for this could only be explained by assuming that a lack of balance in the Y. S. T. engine produces a different effect from lack of balance in others.

Very careful consideration of Herr Schlick's paper of 1901 fails to convince us of the correctness of his position, and it would require much more definite evidence than he adduces to establish the fact that the comparatively large effects shown by the vertical vibrations were due to the greater resistance experienced by one blade of each propeller over that acting on the other blades. The propellers are, no doubt, of high class workmanship, and it is difficult to believe that any such marked difference in pitch or surface in one particular blade of each propeller would exist as is required in Herr Schlick's deductions. To be convincing, his paper should have given careful measurements of pitch and surface, which would have enabled his claims to be investigated numerically. These measurements could very readily have been made at any time the Deutschland was in dry-dock. A slight diminution of pitch in the blades A, referred to in his paper, should have been accompained by a disappearance of the first period vertical vibra-

tion, giving another easy confirmation of his theory, which ascribes the vibration of the Deutschland entirely to what must be considered very bad workmanship on the propeller. The comfort of the ship would also thereby have been increased.

As further supporting the opposite view, that with a well-made propeller the vibrations almost entirely originate in the action of the engine, I may give the following quotation from Mr. Yarrow's 1892 paper:—

"From our experiments we have overwhelming proof that the vibration of a torpedo-boat is precisely the same in extent and character when the screw is on, and the vessel driven through the water, as when it is stationary and the engines simply revolving without doing work, the propeller being removed."

We have every reason to be certain that these large second-period couples, shown by Fig. 2, may vibrate the ship. That a ship responds readily to first-period couples can easily be proved. Thus, Mr. Yarrow, in a three-crank engine, made the moving weights of all three cylinders equal, thereby annulling, almost perfectly, the free force, but leaving the couple. He says, in his 1892 paper:—

"Thus we prevented any vertical movements of the center of gravity of the engines; yet we found no improvement. This clearly indicates that the rocking vibrations are of more importance than the vertical vibrations in triple-expansion engines."

As proving the same, Mr. Macalpine points to the experiments on H. M. S. Powerful and Terrible.*

If first-period couples are so important, why should it be reasonable absolutely to ignore second-period couples?

From the large proportion of the reports one hears, ships having engines on the Y. S. T. plan seem to behave in the manner that the complete theory would lead us to expect.

I will now examine very approximately the unbalanced forces and moments of second and fourth period for the Deutschland.

All the data used here will be found in Engineering for November 23, 1900, and March 22, 1901.

* "Recent Trials of the Machinery of Warships," by Sir Albert John Durston, K. C. B., R. N., M. Inst. C. E., and Henry John Oram, R. N., M. Inst. C. E.; Minutes of the Proceedings of the Institution of Civil Engineers, Vol. CXXXVII, p. 210.

Fig. A, Plate 46, gives the crank angles, and Fig. D the crank centers in millimeters, and feet and inches, and the cylinders worked from each crank.

The size of the engine is:—

Diameter of two H. P. cylinders,	930 mm. =	36.61	in.
" first Int. "	1870 mm. =	73.6	in.
" second Int. "	2640 mm. =	103.9	in.
" two L. P. "	2700 mm. =	106.3	in.
Stroke of pistons,	1850 mm. =	72.8	in.

Connecting rod = 4 cranks.

The weights of the moving parts for cranks II and III were calculated from the general arrangement of engine, and cylinder and crank shaft details, published in Engineering. They can only be looked on as approximate, but the details are given to so large size and so fully dimensioned that an error of more than from 5 to 10 per cent. is improbable. The mass taken—32 tons—is probably a little under the truth. The comparatively small effects from the valve gears are omitted here.

The engine is supposed to be running at 90 revolutions per minute, giving a piston speed of 1092 ft. per minute, which is about that usually maintained in the Atlantic greyhounds.

$$\omega = \frac{90 \times 2\pi}{60} = 9.4248.. \quad \omega^2 = 88.8.$$

m_1 for cranks II and III = 32 tons.

$r = 72.8 \div 12$, feet.

$\frac{m\omega^2 r}{g} = \frac{32 \times 88.8 \times 72.8}{12 \times 32.2} = 268$ tons; the highest inertia force of first period for cranks II and III.

If we take all moments about crank I we have:—

Maximum value of moment for crank II = $268 \times 11.5 = 3082$ ft. tons.
" " " " III = $268 \times 30^{10}/_{12} = 8263$ ft. tons.

We can now determine the other moments and forces, necessary for first-period balance, from Fig. A.

Lay off *oa* to represent to scale 8263 ft. tons.

Draw *ab* parallel to crank IV and produce crank II in *ob*. *ob* then represents the necessary moment for crank II and *ab* that for crank IV.

ob = 3082 ft. tons.
ab = 7900 ft. tons.

(It may be noted that the data we are using is redundant. That it does not give us any trouble is proof that the crank angles have been correctly determined. I need not stop to remark on the data necessary and sufficient to make the required determination in any case. That will be found discussed in some of the papers referred to above.)

The force for crank IV is $\frac{7900}{42\frac{1}{3}} = 186.6$ tons, $42\frac{1}{3}$ being the arm of the moment. (See Fig. D.)

Now draw the force polygon *ocdeo*, Fig. A, by making the sides proportional to the forces for the several cranks, to which they are parallel.

We find *eo* = 186 tons.

Higher period forces and moments.

Only the parts having the same motion as the crosshead, and a part of the connecting rod* give rise to these shorter period forces. The mass of these parts for the Deutschland is about 20 tons for each of the cranks II and III, or $\frac{5}{8}$ of the mass producing first period forces. Light counterbalances on cranks I and IV make the proportionate reduction for these cranks about the same.

Again, the connecting-rod length being four cranks, the coefficients in the infinite series of equation (14) apply to this case. Thus for second period we must multiply the forces and moments for the four cranks in Fig. A by $\frac{5}{8}$ for the reduction of mass, and by .2540, the coefficient in equation (14). For fourth period the multipliers are $\frac{5}{8}$ and .0041.

Second Period.	*Force.*	*Moment.*
Crank I.....	186 × .2540 × $\frac{5}{8}$ = 29.5 tons.	Zero.
Crank II	268 × " " = 43 "	3082 × .2540 × $\frac{5}{8}$ = 489 ft. tons.
Crank III..... ..	268 × " " = 43 "	8263 × " " = 1312 "
Crank IV.....	186.6 × " " = 29.6 "	7900 × " " = 1254 "

Fourth Period.	*Force.*	*Moment.*
Crank I.....	186 × .0041 × $\frac{5}{8}$ = .48 tons.	Zero.
Crank II	268 × " " = .69 "	3082 × .0041 × $\frac{5}{8}$ = 7.9 ft. tons.
Crank III.....	268 × " " = .69 "	8263 × " " = 21.2 "
Crank IV	186.6 × " " = .48 "	7900 × " " = 20.2 "

Figs. B and C, Plate 46, give the crank angles doubled and quadrupled, respectively. The above forces and moments are laid off parallel to the corresponding cranks, as in Fig. A, but in Figs. B and C the polygons do not close, showing that the engine is unbalanced for second and fourth period.

*The proportion of the whole rod to be taken is the ratio which the distance between the center of gravity of the rod and the center of crank-pin bears to the whole length of the rod.

The defect of balance is :—

	Force.	*Moment.*
Second period.....	1.7 tons.	1910 ft. tons.
Fourth period.....	1.71 "	37.2 "

These values are for one engine only.

Second Period. The force is quite unimportant, but the moment is very great.

It is strange that Mr. Schlick should have devoted his elaborate paper of 1900 to showing how this force could be reduced to zero, and have passed over the moment as quite negligible.

The importance of forces and moments of a given value is greater the shorter their period, since the shorter the period the more quickly will energy of vibration be supplied, other things being equal. Just as an engine running fast will give more horse-power from the same steam loads than the same engine running slow. Thus 1,910 ft. tons of second period is as important as a much larger moment of first period.

Fourth Period. The remark just made applies with greater force to this period. Besides, these forces and moments are here estimated for a perfectly rigid foundation. They are due to large masses with an extremely small amplitude of vibration. Whenever the ship responds their values will increase, and we can readily see that in the case of fourth or higher period, where the original amplitude is minute, the response from the ship may be such as to increase the exciting forces and moments many times. Also, as a vibration becomes more rapid, the amplitude, which can be tolerated, diminishes very quickly.

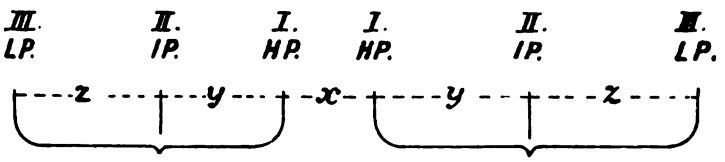
This great increase of amplitude cannot, of course, occur where the original amplitude is comparatively large, as in the first and second periods. The full discussion of the question is very difficult, but we can readily see that forces of high period must not be treated as negligible, because they are relatively small. Experience shows that they give rise to very sensible effects.

2d. Let us now notice an engine proposed by Messrs. Robinson and Sankey in a paper read before the Institute of Naval Architects in 1895.

If we make a three-crank engine with cranks at 120°, and all the masses, m, attached to each crank equal, the first-period force will be zero. For it is

$$\frac{m\omega^2 R}{g}\{\cos\theta + \cos(\theta + 120°) + \cos(\theta - 120°)\}$$

$$= \frac{m\omega^2 R}{g} \times \cos\theta\,(1 + 2\cos 120°) = 0, \text{ since } \cos 120° = -\tfrac{1}{2}$$



1st eng. Fig. 3. 2nd eng.

Continuous bed plate.

But the first-period couple is obviously not zero. Now mount two exactly similar engines of this kind, but right and left hand, on one bed-plate, Fig. 3. Corresponding cylinders, say the H. P., are to be contiguous, so that the cylinder centers, if not equal, may be symmetrically placed. Couple the shafts so that the contiguous cranks of the two engines pass top center at the same time. Then will the center cranks of each engine pass top center together, and also the outer cranks.

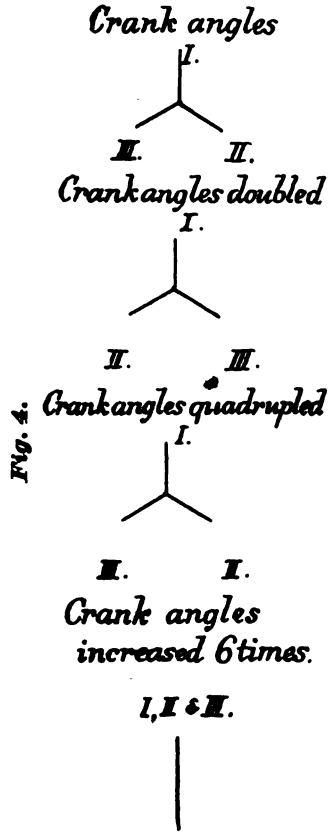


Fig. 4.

First period forces are zero. First period couples from each engine exactly oppose, and are therefore zero for the combination.

That is, the first period balance is exact.

Double the crank angles and we get exactly the same arrangement of cranks, but left-handed. Hence the second period balance is exact.

Quadruple the original crank angles and we get the original arrangement. Hence the fourth period balance is exact.

Increase the original crank angles six times and all the cranks come into one plane. Hence there is the maximum unbalanced force of sixth period. And so on.

These changes are shown in Fig. 4.

If we could bring these engines together till the two contiguous cranks e one, actuating a mass $2m$, the six-cylinder engine would become a five- inder, with balance up to and including the fourth period.*

ust be These engines have found almost no favor, and are not likely to, as they alance is very costly and long. Space is too valuable on board ship. Their Mr. Macalpine lays much stress upon the fact that calculations for balance of the four, five, or six-crank engines, and also for the revolving counterbalances at the ends of the engine, proceed on the assumption that the ship is rigid over the length of the engines, and that the assumption is specially inadmissible when considering the shorter period vibrations; in fact, that the whole question is one of the elasticity of the ship. It is not unnatural that he should notice the point, as the balance of the engine he proposes does not involve this assumption to anything like the same extent, but I cannot think that this point is of more than the most secondary importance.

Before proceeding to consider in detail the Macalpine system of balancing, and in order that we may not omit any system which has a legitimate claim to be considered in relation to the balancing problem, we may give some thought to the steam turbine. The splendid work done by the Honorable C. A. Parsons in the development of this form of motor, and the remarkable success which has attained its application to torpedo vessels in the way of securing extraordinary speeds, has led many enthusiastic members of the profession to believe that this motor is destined ultimately to supplant the reciprocating engine. With a view to give it a fair trial, Messrs. Denny built the excursion steamer King Edward, which has recently been put in service on the river Clyde. Nearly all of the published accounts of trial trips of torpedo-boats have called particular attention to the marked absence of vibration at high speeds, so that this has already been advanced as a strong argument in favor of the adoption of the turbine. In addition to others which have been given, it should be stated at once that the turbine is perfectly balanced, for it would be impossible to run it at the high speeds necessary, unless it were in perfect dynamical balance.

It is to be noted, however, that there are some very serious drawbacks to the steam turbine, as far as it has now been developed, for general use as a marine engine. One is the difficulty of reversing, and, as is well known,

*The text only gives a particular case of the five and six-crank engines, which is ample for the purpose of this paper. For a complete statement of the necessary relations between the moving masses and distances of cylinder centers, see "On the Balancing of the Reciprocating Parts of Engines, Including the Effect of the Connecting Rod." By Prof. W. E. Dalby, M. A., B. Sc., §§ 23 and 24. Trans. Inst. Naval Architects, 1901.

the vessel is driven astern not by the same turbines which give the ahead motion, but by independent ones which, from the necessities of the case, are much smaller and give only a relatively low speed. It has been asserted by many that the recent loss of the Viper, from whose performances so much information was expected, was on account of this lack of full power for reversing. Another objection is the decided lack of economy at very low powers. While this might not apply to the large liners which run at nearly constant speed throughout the voyage, it is a very real objection to use on war vessels which, during most of their cruising, run at about one-tenth of their full power. It is true that Mr. Parsons has proposed to increase the efficiency at low powers by fitting small triple-expansion engines forward of the turbines, with a clutch connection to the main shaft. This proposal, however, can hardly receive much approval, for the reason that it would introduce altogether too much complication of machinery.

I understand, moreover, that while the turbine itself is completely balanced and there is no vibration of the ship forward of the turbines, experience on the King Edward has shown that there is a decided vibration, which increases from amidships aft. In the saloons, especially in the lower or dining saloon, the vibration and noise are very great, and, indeed, at the after end of the saloon the vibration is said to be almost unbearable. In this respect, this vessel is probably the worst on the Clyde. This vibration is partly due to the propellers, for it must be remembered that the speed of revolution is abnormal compared with that of ordinary vessels, and a degree of balance which would be considered perfect in a propeller running at ordinary speed would be inadmissible for these very high-speed cases.

In any event, the steam turbine is at present entirely in the experimental stage, and whatever its future development may be there can be no question that we shall go on building reciprocating engines for a great many years to come. Consequently, from the standpoint of actual practice, the turbine cannot be considered a practical solution of the balancing question.

Let us now consider Mr. Macalpine's solution of the vibration problem, taking largely as a guide his recent paper, read before the Institute of Naval Architects this year in their summer meetings in Glasgow.

We have seen, earlier in this paper, that only partial balance can be attained by direct-connected four-crank engines; and that, though the solution with five and six-crank engines is much more nearly perfect, there are great practical objections to their general adoption. The complete solution, then, is not to be attained by direct-connected engines of the usual type.

There is still one solution with the connecting-rod engine, and it is unique.

Add to the right-hand side of equation (14) an expression—

$$-\frac{m\omega^2 r}{g}(\cos\theta + .2540\cos 2\theta - .0041\cos 4\theta + .00007\cos 6\theta + \text{etc.}). \quad (16)$$

Make $$MR = mr$$

and we at once have

$$F = 0.$$

In the Watson and Billetop balanced engine, this solution is realized with the most complete success. It is a four-cylinder engine. Pairs of cylinders are placed exactly in line, but on opposite sides of the crank shaft. The cranks for opposite cylinders are exactly opposite, so that the motions of the pistons are exactly the same, but in opposite directions. This makes the addition of expression (16) to the corresponding expression in (14), as directed above. In Engineering of July 5, 1901, p. 7, Mr. Billetop writes :—

" During the tests, the engines were not held down by bolts or fixtures of any kind, and a glass filled with water and placed on top of the cylinders did not show the slightest surface tremor when running under those conditions."

FIG 6

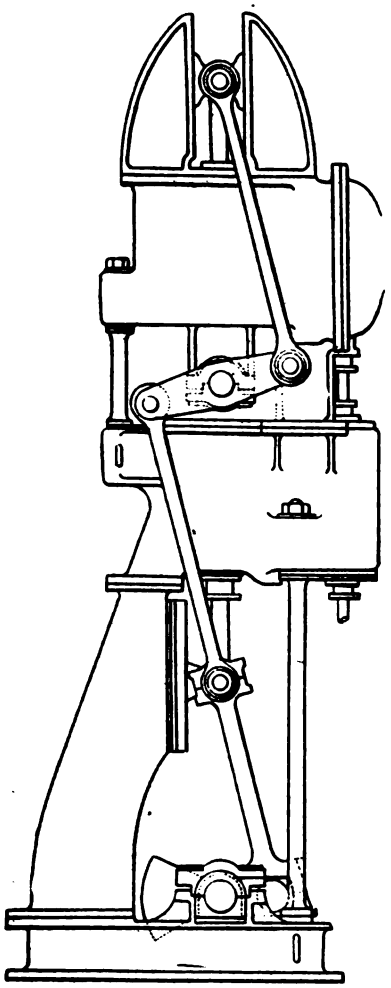


Fig. 6 also shows, roughly, a design of engine in which this solution is realized. The counterbalance on the shaft balances the inertia of the crank and part of the connecting rod. The rest of the lower moving parts are balanced by the oppositely moving upper parts. By making these adjustments correct, the vertical and horizontal balance can be made exact.

Figs. 7 to 18, Plate 43, show a design which Mr. Macalpine gives in his paper, but which he has now improved on. He points out that the motion of the back piston is not *absolutely* the reverse of that of the piston over the shaft. The difference of displacement is, however, never more than a very small fraction of an inch, and I retain the design as a very approximate solution, though I am entirely satisfied that the modification in Figs. 19 to 21, Plate 44, is to be preferred, not only because it gives a perfectly exact solution, but for purely practical questions of design. Figs. 22 to 27, Plate 45, shows another form of exact solution only suitable for merchant ships where there is ample headroom.

In each of these designs a comparatively light counterbalance on the crank exactly balances it and the other parts revolving with it. The horizontal forces are thus balanced. The reciprocating masses, moving oppositely in the pairs of cylinders, can also be adjusted so as to balance. In the designs shown the stroke of piston for each cylinder is equal, so that what is required is to make these oppositely reciprocating masses exactly equal. The calculations for the design, Figs. 7 to 18, Plate 43, I append, to bring out all the details that have to be attended to. I do not give the figures for the other designs, as these given are inserted merely to act as a guide to those calculations which will have to be repeated for each new design as it is made.

But we are left with an unbalanced inertia torsional couple, since the

connected pairs of pistons, etc., do not move in the same vertical line. By placing the cylinders tandem or adding a third cylinder on the other side of the vertical line through the shaft, connecting it also by a lever, this increase of the unbalanced torsional couple is avoided. But these arrangements are not to be commended for shipboard, and I will show that the simpler design is justified by the small effect of the unbalanced torsional couple. This effect will be to vibrate the ship torsionally about a longitudinal axis. We will require to examine carefully the magnitude of this vibration since the admissibility of the engine as a solution of the problem depends on its being exceedingly small. This magnitude is, happily, very easily predicted. We will leave this question till near the end of the paper and give first the necessary adjustments for exact vertical and horizontal balance.

"Cylinders 27 inches, 39 inches, 56 inches, and 80 inches diameter.

"Stroke, 48 inches.

" Boiler pressure, 200 pounds per square inch.

" Designed piston speed, 900 feet per minute.

" Revolutions, 112.5 per minute.

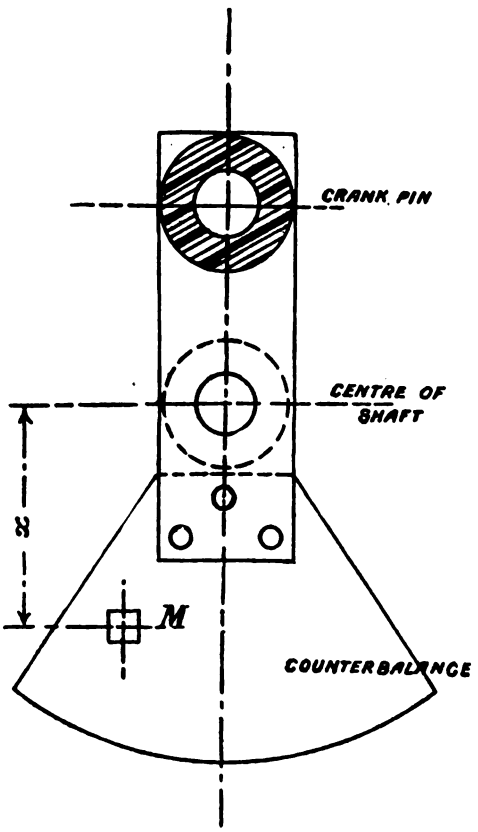
CALCULATIONS FOR BALANCE.

" (*a*) Connecting rod*:—

One connecting rod		= 3,648 lbs.
Length between centers, 9 feet		= 108 in.
Center of gravity of rod from crosshead centre		= 76.38 in.
Part of connecting rod to be taken as revolving with crank pin	$= 3{,}648 \times \frac{76.38}{108}$	= 2,580 lbs.
Part of connecting rod to be taken as reciprocating with crosshead	= 3,648 — 2,580	= 1,068 lbs.

* For the reason of this calculation, see "Analysis of the Inertia Forces of the Moving Parts of an Engine," by the present writer; Engineering, Vol. LXIV, p. 513, remarks after equations (15) and (17); or, Journal of the American Society of Naval Engineers, Vol. IX, p. 513, remarks after equations (83) and (85).

"(*b*) Counterbalancing the parts revolving with the crank :—



	Mass. Lbs.	*Arm for Leverage. Inches.*		*Moment. Inch-pounds*
Part of connecting rod to be taken as revolving with crank pin—from (*a*)	2,580	× 24	=	61,920
Crank pin taken as solid	1,158	× 24	=	27,792
Crank webs on crank pin side of shaft center	2,930	× 14.1	=	41,313
				131,025
Deduct for hole at crank radius	579	× 24	=	13,896
Total amount				117,129
Two counterbalances and parts of crank webs on same side of shaft center*	5,250	× 22.3	=	117,075

"The counterbalances thus exactly balance the other parts revolving with the crank, as the inertia forces are equal and oppositely directed in the same plane.

"(*c*) Reduced mass to be taken for levers.

"As all parts of the main levers have not the full 48-in. travel, they must be taken at a reduced weight, the moment of each end being reckoned from the rocking center, and the arm being measured parallel to the center line of the lever, similarly to the calculation for the counterbalances in (*b*), above.

	Crank End of Lever.	*Outer End of Lever.*
Length between rocking center and end centers	39 in.	39 in.
Mass	2,482 lbs.	2,123 lbs.
Moment	54,604 in.-lbs.	44,160 in.-lbs.
Arm = moment ÷ mass	22 in.	20.8 in.
Reduced mass	$\frac{2,482 \times 22}{39}$ = 1,400 lbs.	$\frac{2,123 \times 20.8}{39}$ = 1,132 lbs.

* Obviously, in counting the moment for the counterbalances, the sum to be taken is ΣMx, where M is any small portion of the mass and x its distance from the shaft center, measured parallel to the crank radius. See above sketch.

"(*d*) The parts moving with the crosshead of the high-pressure cylinder must now balance those of the second intermediate pressure; similarly, the corresponding parts of the first intermediate pressure and low pressure must balance, as in the following table:—

	High Pressure.	2nd Intermediate Pressure.	1st Intermediate Pressure.	Low Pressure.
	Lbs.	*Lbs.*	*Lbs.*	*Lbs.*
Piston	662	2,329	1,416	3,083
Piston rod	1,180	1,428	1,180	1,428
Guide block	1,425	689	1,425	689
Part of connecting rod. See (*a*)	1,068	—	1,068	—
Links to levers	1,067	1,224	1,067	1,224
Reduced mass of levers	1,400	1,132	1,400	1,132
Total	6,802	6,802	7,556	7,556

"The vertical forces from these parts are now exactly balanced, and are zero.* The piston rods and crossheads of the high pressure and first intermediate pressure are shown bored out, merely to call attention to the fact that the balance may be obtained by *deducting* weight as well as by the less costly way of adding weight, say to the pistons, where necessary.

"In the design shown, it has not been necessary to depart from the usual proportions of the parts to effect a balance.

"(*e*) The parts of the main levers, and links at their ends, have a small transverse movement, principally of second period; but the forces from this are also balanced thus:—

	Crank end.	*Outer end.*
Mass of links	1,067 lbs.	1,224 lbs.
Length of links	55 in.	55 in.
Distance of center of gravity of links from upper ends	27.5 in	35 in.
Reduced mass of links for estimating horizontal forces	$\frac{1,067 \times 27.5}{55}$ = 533.5 lbs.	$\frac{1,224 \times 35.}{55}$ = 779 lbs.
Reduced mass of levers from (*c*)	= 1,400 lbs.	= 1,132 lbs
Total	1,933.5 lbs.	1,911 lbs.

*Approximately, for design Figs. 7 to 18. Exactly, for the other designs.

"To produce this balance, the extremity of the outer end of the lever has been thickened to 5½ inches, whereas 4 inches is ample for strength. By a very small further addition, the totals 1,933.5 and 1,911 could be made to agree exactly; but they are left, in order to indicate that an infinitesimal residuum need not be taken account of.

"(*f*) Mode of procedure.

"If the following sequence be observed in making drawings and calculations, no further delay need be incurred in the design than the time required to make a few simple calculations.

"(1) After the crank shaft and connecting rod are drawn, calculate and draw the counterbalance of the requisite size.

"(2) After the main levers and their links are drawn, make any slight addition required to one or other end of the levers to produce horizontal balance, as in (*e*).

"(3) After all the other moving parts are drawn, add to one or other of the balancing pistons sufficient to make exact balance, as in (*d*).

"This also suggests the proper mode of procedure in building the engine. A little metal should be left on each piston till all the other moving parts are finished and weighed, after which the requisite amounts of metal should be taken off to produce the exact required balance.

"(*g*) The balance of the valve gears is obtained in precisely the same way as the foregoing. But it would be ample to make—

Reduced mass of ahead eccentric.....
+ ahead strap and rod.....
+ (say) ½ link.....
+ ½ drag-rod.....
+ valve-spindle .....
+ reduced mass of crank end of levers....
+ high-pressure valve..... ... .

=

Reduced mass of outer end of levers
+ crosshead
+ valve spindles
+ 2d intermediate pressure valves.

"And, similarly, for the first intermediate pressure and low pressure gear."

Obtaining an exact vertical and horizontal balance for all periods will prevent those short vibrations or tremors which are not only annoying, but must make the correct aiming of a gun much more difficult.

The alternative and modified designs, Figs. 16 to 21, show the clearances of the steam passages in the cylinders over the cranks reduced to a minimum; the engine will thus be brought up to the highest efficiency of any modern marine engine. The counterbalances marked "*G*," Fig. 16, are added to bring the inertia of the valves of these cylinders into the plane of the valve levers.

Mr. Macalpine discusses the various points of the designs, Figs. 7 to 18 and 22 to 27, very fully, but in this we need hardly follow him. A careful examination of the design will show that practical questions of construction, accessibility and overhaul, have been fully attended to. But as the design, Figs. 19 to 21, is now put forward for the first time, I may give a few notes to show wherein it has even been improved in these respects.

The bearing surfaces in this and the merchant engine design have all been increased. Though the surfaces in the earlier design seem ample, the increase puts the long running of the engine without readjustment beyond question.

By bringing the main guides to the outsides, their accessibility is considerably increased, and it also allows the levers, though raised, to be kept closer together with their center bearings outside them and in easy reach. Making a center front column leaves the engine perfectly open over the main bearings, so that not only these, but also the main crossheads, front links, and center lever bearings, can be easily got at, even when the engine is running. To make this easier at the after end the after back column of the second I. P. cylinder is moved a few inches aft of the center of the corresponding main bearing. As strength is easily provided, this shift may be made without fear. Also, to give better headroom above the intermediate platform, the H. P. and first I. P. valves are worked from above. This is also improved by dispensing with the jaw guides of these gears, as shown in the view of the valve gear, Fig. 21.

The extra length of back piston rod, required by the modification of the design, is 25 inches, but this weight is almost entirely saved by the shortening of the links. In any case, its inertia at full speed only increases the pressure on the main bearings by 2½ pounds per square inch, so that the addition is unimportant.

The travel of the levers above and below the horizontal position is made unequal. This in no way affects the balance, but by preventing the lever and links from drawing so nearly into line when the crank is at the top makes the adjustment of the whole connection less delicate. The links have also been so designed that they can be kept exactly to length. The levers can readily be withdrawn to the back. The back links and glands of back cylinders are in easy reach.

The crossties are attached so that the expansion of the cylinders will have no sensible effect.

Some of the weight saved by bringing down the back cylinders is added to the front columns and some to the front of the bed-plate, so that the total is not much altered.

The front cylinders are held from fore-and-aft movement by a jaw cast on the L. P. cylinder, which holds, by sliding contact only and without bolting, the foot for the center back column of the front cylinders.

The front of the engine may now be toward the center of the ship, as is usually most convenient; and the hatches, instead of coming close together, will leave a wide passage on deck between them. The hatches do not now require to come much beyond the center of the back cylinders, which will leave these openings of about the ordinary width, and give broad side passages on deck.

The H. P. and first I. P. valves draw up, those of the second I. P. and L. P. draw down. This not only suits the design, but also the narrower hatches.

Is it admissible to introduce the lever into high-speed engines? So far as the valve gear is concerned there are ample precedents, and consequently there can be no question. Not only has the Stephenson link been used with an almost identical arrangement of lever, but we have in the Marshall gear quite as indirect a connection, and in the Heusinger-von-Waldegg gear one much more indirect. But these are good gears, and only require to be well and carefully designed to give great satisfaction.

In regard to the levers connecting the back cylinders, Mr. Macalpine gives an elaborate defence of their introduction. In Britain, the beam engine is rarely or never seen on board ships, but we cannot cross our ferries or sail up our broad rivers, or take a summer trip up Long Island Sound without being for the time shipmates with one, thus having ample time to become acquainted with their good qualities. The piston speed—and it is principally piston speed that counts—runs sometimes as high as 1,000 feet per minute, and the beam works often to a great angle. There is no smoother working engine made, and it is the most economical in the use of lubricants of any engine known; and the chief engineer may often be seen sitting in his easy chair, the picture of unconcern.

Mr. Macalpine quotes our experience, and, having lived for six years with us—having had the misfortune to just miss being turned into a good citizen of our Republic—ought to know. The beam engine forming part of our earliest recollections and of our maturest experience, we hardly need the arguments he adduces to convince us. He speaks of the number of joints between the piston and crank. He has precisely the same number as in our beam engines, and we experience no trouble. It is all a question of bearing surface.

In our original Chicago we had, to the writer's mind, the worst designed

needs explanation further than to point out that the moving parts of the pumps are also balanced.

As noted earlier, it is not sought to balance the torsional couple about

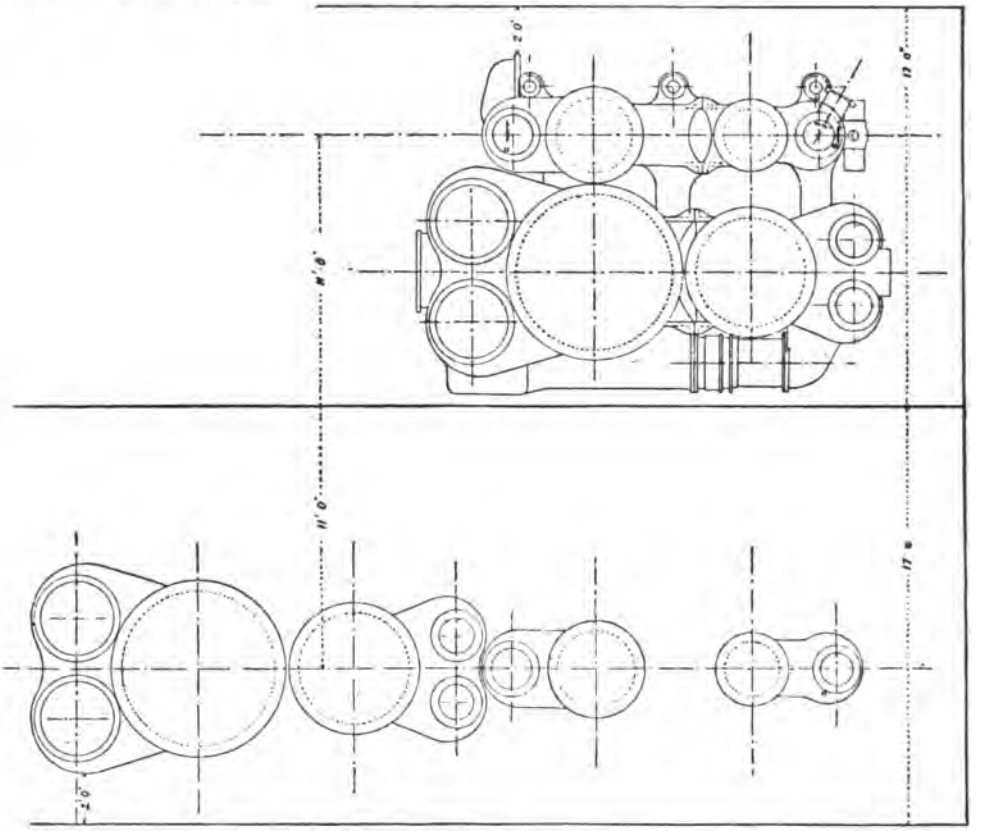


FIG. 28.

the crank shaft, as Mr. Macalpine has shown that the effect in vibrating the ship is excessively small. I will only transcribe the results, referring to the original paper for the full details of the calculation.

He first shows that sensible elastic torsional vibration will not occur at all.

The ship is composed of parts such as the skin and decks which give torsional rigidity, together with machinery and woodwork, etc., which do not add sensibly to the elastic reaction. These latter parts, together with the cargo, add to the moment of inertia about the axis of vibration, and thus lengthen the period of torsional vibration. From actual figures supplied by Prof. F. P. Purvis, Mr. Macalpine concludes that the total moment of inertia of a ship will range from a higher limit of about three times that due to the

parts giving torsional rigidity, to less than twice; the figures for warships being nearer the lower limit. But he calculates the result for an increase of two, three, and four times, showing that even the latter high value would not affect his argument for the absence of elastic torsional vibrations.

He then gives the ordinary investigation for the period of torsional vibration of a circular tube or bar, the mode of vibration being that with two nodes, as these are the slowest torsional vibrations into which a ship would be thrown by couples applied at the center by an engine placed there.

If T = the time of a complete vibration in seconds;
l = the length of the tube in feet;
n = the increase of moment of inertia (2, 3, and 4 times) as explained above.
ρ = the weight per cubic foot of the elastic material of the tube.
μ = modulus of rigidity.
g = the intensity of gravity = 32.2.

Then,

$$T = l\sqrt{\frac{n\rho}{g\mu}}, \tag{17}$$

when there are two nodes. The speed of vibration is independent of the diameter of the tube.

I will quote the numerical results from Mr. Macalpine's paper, making the requisite changes in the numbering of his equations.

"ρ for steel = 500 lbs. per cubic foot.
μ " = 12×10^6 lbs. per square inch.
= $12 \times 144 \times 10^6$ lbs. per square foot.
g = 32.2.

"For a tube (ship) 400 ft. long we get from equation (17)

$$"T = 400\sqrt{n}\sqrt{\frac{500}{32.2 \times 12 \times 144 \times 10^6}} = .03792\sqrt{n}.$$

If N = the number of vibrations per minute—

$$"N = \frac{60}{.03792\sqrt{n}} = \frac{1582}{\sqrt{n}}. \tag{18}$$

"In a ship 400 ft. long, 112.5 may be taken as near the highest number of revolutions of any engine which will be put in it, even if it is a warship. The number of the synchronizing period then is—

$$"S = \frac{1{,}582}{112.5\sqrt{n}}. \qquad (19)$$

"We thus get the following table:—

" n = multiplier for moment of inertia.....	2	3	4
" N = number of vibrations per minute, equation (18)	1,119	913	791
" S = number of synchronizing period, equation (19)	9.9	8.1	7.0"

Mr. Macalpine finds that for several other sections the results do not change much from the above. What is required to make a torsionally stiff structure is to so stiffen it that the sections can change shape but very slightly when the twisting couple is applied. This is amply done in a ship by means of the frames, beams, and bulkheads; so we may be sure the above result does not differ greatly from the truth for a ship. Especially is this the case since the time of one vibration lengthens in the same proportion as the square root of the elastic reaction diminishes. We see, then, that the proportionate change in the period is much smaller than that of the elastic reaction.

The cylinders being arranged in pairs athwartship, the intensities of the torsion couples due to inertia are changed from those of the ordinary design.

Mr. Macalpine calculates the increase from that of second period upward over what it would be for the same two cylinders placed over the shaft; that is, the increase due to the displacement athwartship. The results are:—

Period.	*Percentage increase.*
2d.....	25.7
3d.....	0
4th.....	7
5th.....	0
6th.....	3.4
7th.....	0
etc.	etc.

In his paper it is shown that the synchronizing period will be somewhere in the neighborhood of the seventh to the tenth, and the results just given show that there will be no increase of the tendency to set up elastic torsional vibrations of these periods in this engine over what we have in the

ordinary four-crank engine. Hence, we need not fear these vibrations, as they do not occur with the ordinary engine.

There is a very large unbalanced first-period inertia torsional couple produced by this engine, which scarcely exists in the direct-connected engine. It will produce no elastic torsional vibrations, as it is only applied from a seventh to a tenth as quick as the slowest of these vibrations, but it will rotate the ship as a rigid body. As, for instance, if we suspended a watch by a thread with its face horizontal, the vibration of the balance wheel would produce a very small vibration of the whole watch. These are not elastic vibrations, but vibrations of a rigid body. Their extent can readily be determined if we know the angular swing and moment of inertia of the balance wheel, and also the moment of inertia of the whole watch; for the moment of momentum of the whole watch, including the balance wheel, must remain zero. This is well known to everyone, and it holds also with the ship and engine.

Now suppose the connecting rod infinite in length, the velocity of the pistons will be greatest at the center of their travel.

Using the same symbols as before, the velocity at the middle of the stroke is ωr, and the momentum for the moving masses of one cylinder is $m\omega r$.

To simplify the explanation, suppose we are dealing with the H. P. and second I. P. which are connected by a lever. m is the same for each as the counterbalance, and the parts it balances produce no torsional couple, and so may be omitted. $m\omega r$ is, then, the same for both cylinders, the centers of which stand at a distance D from one another. Take the moment of momentum about a point in the center line of the H. P. cylinder. The moment for the H. P. itself is then zero as the lever arm is zero. That for the second I. P. is $m\omega rD$. If we took the moment about a point in the plane of the cylinder center lines, at a distance a from that of the H. P., we would have:—

$$\begin{aligned} \text{For the H. P. moment} &= m\omega r \times a \\ \text{For the second I. P. moment} &= m\omega r \times (D-a) \\ \text{Total moment} &= m\omega r \times D. \end{aligned}$$

Thus the value of the moment does not change with the position of the point around which we take it.

If ρ is the distance of any small mass M of the ship from the axis around which it revolves torsionally, and φ_1 is the greatest angle by which it departs from its undisturbed position, the linear displacement of M from its mean position has a maximum value $\rho\varphi_1$, and the velocity with which it passes through its mean position is $\rho\varphi_1\omega$, since the vibration is of the same period

as the engine revolution. The greatest momentum is, then, $M\rho\varphi_1\omega$, and the moment of momentum $M\rho\varphi_1\omega \times \rho = M\rho^2\varphi_1\omega$. For the whole ship this moment of momentum becomes $\varphi_1\omega\Sigma M\rho^2 = \varphi_1\omega I$, where I is the moment of inertia of the ship about the longitudinal axis for these vibrations. If the total moment of momentum of engine and ship is to be zero, they must oppose one another, and we must have

$$\varphi_1\omega I = m\omega rD$$

or

$$\varphi_1 = \frac{mrD}{I}. \tag{20}$$

The linear amplitude of this vibration will be greater the further we are from the axis. At the side of the ship it will be

$$\varphi_1 b = \frac{mrDb}{I}, \tag{21}$$

where b is the half breadth of the ship.

Mr. Macalpine says that the time of roll of the ship and the action of the water will have no effect on this vibration. While he is right in the first part, I think he is mistaken as to the action of the water. If the sections of the ship were parts of circles in the immersed portions, the ship might vibrate torsionally with almost no disturbance of the water. But in a very full ship a considerable movement of the water would result, and still more if bilge keels were fitted. This action would be roughly equivalent to an increase of the moment of inertia of the ship and thus would diminish, I believe, often to a very considerable extent, this vibration.

Now, from (*d*), p. 25, we get

m for H. P. and 2d I. P. cylinders = 6802 lbs. = 3.04 tons.
m for 1st I. P. and L. P. cylinders = 7556 lbs. = 3.38 tons.

The velocity of the H. P. piston (connecting rod infinite) when the forward crank is θ past top stroke is

$$\omega r \sin\theta$$

and the moment of momentum for the parts attached to this crank,

$$3.04\ \omega rD \sin\theta.$$

Similarly, since the after crank leads by 90°, its moment of momentum at the same instant is

$$3.38\ \omega r D \cos\theta.$$

The total moment of momentum for both cranks is—

$$\omega r D\,(3.04 \sin\theta + 3.38 \cos\theta)$$
$$= (3.04^2 + 3.38^2)^{\frac{1}{2}} \times \omega r D \sin(\theta + \alpha).$$

Hence in using equations (20) and (21), instead of m we may put $(3.04^2 + 3.38^2)^{\frac{1}{2}}$.

In twin screw ships this effect will at times be doubled, and we will use $2\,(3.4^2 + 3.38^2)^{\frac{1}{2}} = 9.08$ tons.

Mr. Macalpine calculates the rigid body vibration which twin engines, Figs. 7 to 18, would give the U. S. S. Alabama. Her engines are of about the same power.

Displacement load (draught) = 11,562 tons.

Radius of gyration he guesses as = 30 feet, and it cannot be greatly different.

Then $I = 11562 \times 30^2$

b = about 36 feet = 36×12 ins.

m = 9.08 tons, from above

r = 2 ft.

D = 6 ft. (actually 5 ft. 10 ins.)

Then from equations (20) and (21)

$$\varphi_1 = \frac{9.08 \times 2 \times 6}{11{,}562 \times 30^2} = 1.05 \times 10^{-5}$$

$\varphi_1 b = 1.05 \times 10^{-5} \times 36 \times 12 = .0045$ inch.

Total movement at side of ship = $.0045 \times 2 = .009$ inch or less than 1/100 of an inch.

To force home the smallness of this movement he calculates the apparent movement of the sight of a gun on a target three miles distant. This evidently is—

$$\frac{1}{100} \times \frac{3 \times 5280}{36} = 4.4 \text{ inches.}$$

He next shows that the effects of the second and higher period forces in producing rigid body vibrations will be exceedingly small, even in comparison to that of the very small first-period vibration just calculated.

Will these effects be increased or diminished as we change the size of the ship? Let us compare two cases where every dimension of ship A is twice that of ship B, machinery as well as ship; boiler pressure and piston speed to remain the same.

Displacement	Ship B =	$\frac{1}{8}$	of that of Ship A.
Radius of gyration	"	$\frac{1}{2}$	"
I (= Disp. × radius of gyrat.^{2})	"	$\frac{1}{32}$	"
r, crank radius	"	$\frac{1}{2}$	"
D, cyl. centers athwartship	"	$\frac{1}{2}$	"
m, moving masses (linear dimen.^{3})	"	$\frac{1}{8}$	"
Wetted surface	"	$\frac{1}{4}$	"
I. H. P. (as cylinder areas)	"	$\frac{1}{4}$	"
Grate and heating surface	"	$\frac{1}{4}$	"

Hence φ_1 for ship B

$$= \frac{\frac{1}{8}m \times \frac{1}{2}r \times \frac{1}{2}D}{\frac{1}{32}I} = \frac{mrD}{I} = \varphi_1 \text{ for ship A.}$$

Thus φ_1 is not altered, and the I. H. P. per square foot of wetted surface also remains the same for both ships.

Possibly the small ship would have its scantlings rather lighter than in exact proportion to the dimensions of the two ships as supposed, in which case φ_1 for ship B would be a trifle larger than φ_1 for ship A.

The total linear movement, $2\varphi_1 b$, would therefore be little more than half in ship B what it is in ship A.

In very high-powered small warships the I. H. P. per square foot of wetted surface might be four, five, or six times as great as in large warships, but an increase such as this would not bring up $2\varphi_1 b$ to more than a very minute value, especially as in the very high-powered boat the engine would be pressed to its greatest limit of piston speed, and the scantlings kept as light as possible—that is, the power would be taken out of as small and as light an engine as practicable, which both go to reduce φ_1. Also the larger φ_1 is, the more powerful will be the action of the water in opposing the vibration. Thus, even in this, the worst case, the solution is still quite satisfactory.

Macalpine finally refers to twin screws with unbalanced engines as forming a much worse case, in which there is usually no noticeable effect and never one of serious magnitude.

In support of this statement of his, and as strongly supporting his whole contention, I may add another quotation from Mr. Yarrow's celebrated 1892 paper:—

"As a further proof that the vibration is due to the machinery, I may mention that two years ago I made a passage to the United States in one of the very fast twin-screw steamers. I selected a berth in the central portion of the vessel, thinking it a good position for comfort, but the vibration was found to be so excessive that after five days it was scarcely bearable to the passengers whose berths, like my own, were situated at the points of greatest vibration. The vibration was found to vary periodically. When the two low-pressure pistons were descending at the same time it was excessive; but when one low-pressure piston was ascending and the other descending it was entirely neutralized."

Here we have the whole matter brought to a crucial test. Mr. Yarrow's berth being at the center of the vessel was not only at the point of maximum vertical, but also of maximum torsional vibration. No doubt the berth was at the outside of the vessel where the movement from a given angle of torsion would be greatest. The engines were exceedingly far from being balanced, for when acting together they gave rise to vertical vibrations "scarcely bearable." When the vertical forces were exactly counteracting one another, realizing precisely the effect of Macalpine's proposed engine, but with a greatly exaggerated leverage between the opposing forces, the vibration was "*entirely* neutralized." The evidence could not be stronger or freer from doubt; and it enhances its value a thousand fold that it was given by an engineer of the highest ability who had been specially studying and experimenting on this very subject for years previously to 1890. If there had been a sensible torsional vibration it would no doubt have been noted. The conditions were entirely favorable to their being set up. The engines would usually run so nearly at the same speed that elastic torsional vibrations would have ample time to reach their maximum development before the favorable phase passed. The rigid body rotation requires no time to grow; it appears at once of such amplitude as will keep the moment of momentum always zero. Yet none was noted though the unbalanced forces and the lever arm between the engines was so large.

In what immediately precedes, it has been stated that with the ordinary engine elastic torsional vibrations do not occur. It may be objected that Herr Schlick, in his 1895 paper, before the Institution of Naval Architects, speaks

of elastic torsional vibrations, giving formulas and diagrams. He does not mention the length of ship on which his observations were made, but judging from the engine revolutions (150) one is led to conclude it was not a very large ship. His formula, which is equivalent to equation (17), would probably have given not less than 1,500 vibrations per minute. He omits to give a numerical example. But his diagrams of torsional (?) vibrations are marked 150 per minute. Mr. Mallock, in the discussion, calls his attention to the great discrepancy between the calculated and observed periods. Herr Schlick's reply is entirely unsatisfactory. He says the difference is due to the small torsional rigidity of the steel hull. "Every shipbuilder," he says, "knows that the rigidity of a ship's hull against torsion is relatively small, . . ." To reduce torsional vibrations from 1,500 to 150 per minute would require a reduction of torsional rigidity in the proportion $\left[\frac{150}{1500}\right]^2 = \frac{1}{100}$!

As the ship was a German war vessel, it would have a steel deck, and, consequently, a torsional rigidity more nearly 80 or 90 per cent. of the corresponding steel tube. I have little doubt that what Herr Schlick observed was the first period rigid body rotation spoken of above.

I regret very much that in this discussion I have been compelled to point out what I consider errors in Herr Schlick's papers, and I do not wish in any way to detract from the great credit which is due him for the original work he has done in the study of hull vibration and engine balancing. I am sure that if he were present he would appreciate that these criticisms are really a tribute to the prominent position which he occupies in connection with this subject, and that they simply represent an honest difference of opinion. That there should be errors in observations and deductions connected with such a complicated subject, is no cause for surprise, and does not, in my judgment, in any way detract from the credit due him.

In what I have written I have tried to include all that has been carried out practically in engine balancing, which can be regarded as any approach toward the solution of the problem. Of the theory I have given enough to exhibit the exact nature of the problem, and have shown that the indications of theory are fully borne out by the consequent phenomena of ship vibrations. I have shown that the old counterbalancing took much too narrow a view. It was adopted long before the exact nature of the forces had been fully investigated, and it can only be looked on as a first approximation to a solution. The discussion has also, I believe, made it clear that the Y. S. T. system is only a partial solution of the balancing problem, and that while it may annul the unbalanced horizontal forces, the effects of which never gave rise to very special

remark, it gives an engine in which some of the forces left unbalanced exert very nearly their highest possible effect. With this view of the system, it would have been out of place in an already very long paper to give the numerous methods, algebraic and semi-graphical, which have been proposed to determine crank angles and moving masses. The engine is costly, heavy, and on account of the calculations required, troublesome to design. When completed it is only a partial solution of the problem, and for the conditions of the fast vessels and high-speed engines of to-day the solution is far from sufficient.

I have shown as fully as I thought necessary how five and six-crank engines must be adjusted to produce a balance up to the fourth period. That this engine, occupying much valuable space, complex and costly, should have been introduced, even in a few cases, shows how pressing is the necessity for a more complete solution of the balancing problem.

As I stated in the beginning, it is my belief that the Macalpine system is the best solution of the problem of engine balancing to prevent hull vibration that has thus far appeared. For this reason I have placed in strong contrast the features of this and the Y. S. T. system, inasmuch as the latter has been applied in more cases than any other system which claimed to give a reasonably complete solution of the problem.

The discussion has aimed to show that the Macalpine system considers fully the dynamical problem, and gives a complete solution ; that it makes no unwarrantable, even if approximate, assumptions. It has been shown that the lack of complete torsional balance leads to very slight response from the ship, and that this single omission to secure absolutely perfect balance is fully justified, as the advantages gained through the admission of this slight lack of balance very much more than offset the disadvantages. It is to be particularly remarked that the engine which thus realizes the perfect solution of the balancing problem is a perfectly practicable one. It is of a type in common use, and with the performances of which we are perfectly familiar, to which are now added, however, the necessary condition for the solution of this great problem. It has the additional advantages of great compactness, simplicity, and accessibility.

If the discussion which has been given should, as I hope, prove conclusive, I trust that the gentleman who has encouraged the preparation of this paper, and who, if I am correctly informed as to his identity, is fully able to test the Macalpine system, will do so. After a most careful study I am convinced that the system is correct, and I believe the results will amply compensate him for having given it an opportunity for demonstration.

DISCUSSION.

THE CHAIRMAN :—Gentlemen, now that these papers are before you, I am quite sure you will sympathize with the feeling of one of the Committee, who suggested that hereafter anyone who offers a prize for any competition should offer double the amount to the Committee for examination of the papers. [Laughter.] I am sure that the author of the first prize paper will also sympathize with the feeling that I have, that had I known who was the author of the second prize paper I would have been in doubt as to who should have been given the prize when I read the extraordinary praise of the old beam engine. Of course we know, those of us who are familiar with the old-fashioned machine, which is supposed to be now relegated to the past, that it is quite a superior machine after all ; but we do not often expect so high an authority as Admiral Melville to praise it directly, as he has in this paper. The paper is now before you for discussion, and I trust that you will give it the attention that this subject deserves. I trust that all the cosines and Greek letters that are in these papers will not deter you from discussing the question of balancing marine engines. Some of us cannot use all those formulæ with correctness, but there is certainly something in these papers which deserves particular attention, and I trust you will not hesitate to give them attention.

MR. WALTER M. MCFARLAND, *Member of Council*:—I think the College Professors are the only people, besides the authors of the papers, competent to discuss them. I frankly admit that I am not. I said that from the beginning. I believe that these two papers are of extremely great value, and I believe it will very well repay every member of the Society to study them with the utmost care, because I believe there is an enormous amount of the most valuable information in them. Of course, I am not so directly connected with the work of this Society as I was once, and my time is taken up in studying other things, chasing the nimble dollar in other ways, and I expect that a good many others find that, in the practical work of their profession, it is a good deal easier to look up a formula in a book, or, when such authorities as Mr. Taylor recommend a certain thing, or any other of our good friends in whom we have confidence, we feel like taking what is recommended on trust and going ahead. We don't feel like going back to the beginning.

MR. J. C. KAFER, *Member of Council*:—I think that the thanks of this Society, and not only of this Society but of the whole engineering world, should be given to these gentlemen for the elaborate papers they have given.

THE CHAIRMAN :—What about the gentleman who offered the prize?

MR. KAFER :—He is well paid. As one of the gentlemen who examined the papers said, "Why the author did all that work for a prize of $100, I don't know. I would not have done it for a thousand." I think the gentleman who offered the prize got most excellent returns for his investment. He got a most elaborate essay in the first prize paper and also in the paper which took the second prize. I had a little discussion with this gentleman in reference to balancing engines, and, in talking about it, he said it was a difficult matter for a person looking over these papers to tell which should have the prize, but, in looking over the requirements, then it became evident that the prize paper was the one that was selected, and there was no question about that; and, after hearing both papers and scanning them, I think the Committee very justly gave the prize to Mr. Taylor, who, I think, stands at the head and front of engine balancing. I do not know of another man in the world who can compare with him. [Applause.] When you take up this subject, you take up an infinite number of technical questions, practical and theoretical. I remember very well, in years gone by, before Mr. Taylor knew what an engine was, there came up a question in reference to calculating the inertia of the reciprocating parts of an engine, and one of Mr. McFarland's intimate friends, who is now associated with him, began, in a peculiarly sulphurous tone of voice to exclaim why he did not learn that before, and although a case in point had occurred on board of a ship on which he had served as an engineer, he never learned the real reason of the thump in the engine until he began making the actual calculations for inertia of reciprocating parts. So that, although these demonstrations here presented to us are demonstrations from a theoretical point of view, they can all be carried out practically, so that we can use them and make some absolute practical use of them in the balancing of the engine and getting rid of the "thump." I would like, therefore, to express the thanks of this Society, when the proper time comes, to these gentlemen, for their very elaborate papers, and especially so to Mr. Taylor for his.

MR. MCFARLAND :—I second that motion.

MR. KAFER :—I hope someone else will have something to say on this subject.

Mr. W. D. FORBES, *Member*:—Mr. Chairman and gentlemen, I am not a college professor, and I am a great failure in chasing X's; I never catch the right one, somehow or other, so I do not go into this paper mathematically. But I want to go into it just a little way on the purely practical side. The suggestion of the Admiral in the last paper was, I believe, anticipated by me sometime ago. I have looked at the drawing, which he shows here, with considerable attention. My position in engineering is precisely that of a cipher in mathematics—no good at all of itself; only of some value when placed alongside of a figure. I simply produce something on somebody else's ideas. But I find, in following out the ideas of people, in following out those of my good friend, Col. Stevens, who gave the prize for this paper, that my work is important; that is to say, the purely mechanical work. We

have gone from slow movements to very rapid ones, and I find that it is absolutely impossible to do the ordinary grade of work which is good enough for ordinary rotative speeds, and get anything like satisfactory results from it. A few days ago we had an engine—we make them to run up to about 1,200 revolutions, the fastest commercial reciprocating engines—which was only up to about 600. The balance seemed to be all right, but it was perfectly unsatisfactory in its action. On investigation, I found it entirely due to the lack of thorough mechanical production. The crank-pin was not round, although the micrometer showed only a very small fraction of an inch out of round. I found that the fit of the piston would make that engine run very smoothly or very unsatisfactorily. Now, while we can chase these X's and catch them and go down and produce the engine, unless the mechanical work is far superior to that which is generally put on engines, I do not believe that these high rotative speeds would be satisfactory. I simply want to call attention to that fact, that we who are simply ciphers in the engineering world have to combat that, and when we get the engine well made, then the question of balance becomes very much less difficult to maintain at first. But, of course, when it wears and gets out of true, then the whole trouble comes back and we have to refer to the mathematical part, and for that I earnestly hope that my good friend, Professor Peabody, and some others here, will aid us in this X chasing, and let us know something more.

PROFESSOR CECIL H. PEABODY, *Member of Council*:—Mr. Chairman, I do not rise in response to the challenge. It is not necessary for me to add to the praise which has already been given to these articles. The feature which most attracted my admiration in the prize article is the very thorough and complete manner in which the author has worked out the problem so that we may have certain definite results. There is one item I would like to call attention to, and that is the author has used an approximate method; doubtless it may be demonstrated that a good enough approximation may very commonly be carried to a degree where further trouble is not worth while. I believe, however, that approximate methods and graphical methods both, for these investigations, are entirely unnecessary, provided you can get somebody else to do some work for you before you start. [Laughter.] That, however, I think can be done and is done. No one is at all concerned as to how much work it costs someone else to make a table of logarithms provided he can get it and use it. If simple tables are made in a proper form, the accelerations for reciprocating revolving weights can be easily obtained. I call your attention to the discussion given by Professor Lanza upon a paper by Professor George I. Alden, in the Seventh Volume of the Proceedings of the American Society of Mechanical Engineers in 1886, page 181, in which discussion Professor Lanza gives some tables of this sort—tables of the same sort we are accustomed to use. This, of course, is only discussion of method, and I believe that while a graphical method always seems very attractive to a person who is not accustomed to using mathematics all the time, that if we can get this work properly systematized, it can be made even easier than by the use of the graphical

method. Of course, in Mr. Taylor's paper the results which he gives in the diagrams (which, of course, one cannot otherwise than accept on the face as being correct) are so arranged as to permit of their use in very much the same way as tables.

PROFESSOR WILLIAM F. DURAND, *Member of Council*:—I think nearly all admit that no one, whether he be a professor or somebody else, is qualified to intelligently discuss the papers, without taking time for very careful consideration. The papers are so exhaustive and cover such a wide ground that a cursory glance through them or hearing them read is quite inadequate to give that hold on the subject which would furnish proper grounds for any really valuable discussion. There has been so much said already in praise of the papers that perhaps nothing more is needed on that point, but I do feel, in common with the others who have spoken, that we, as a Society, are to be congratulated on having presented at our annual meeting these two papers. I think it is a fact in which we may properly take pride. I think that the donor of the prize may well take much credit unto himself for having been the occasion of having brought out these two papers. One especial point which occurs to me is in regard to the exhaustive character of these papers. All who have followed the literature on the subject know that in the last ten or twelve years there has really been a tremendous output on this particular subject, and if a person would start in to-day simply wishing to look it up, he would find it a very serious proposition to wade through the mass of matter that has been written and printed bearing on this particular subject. Now, while I would not say, and I am sure that the authors of the papers would not maintain, that, in their two papers, the last word has been said on balancing, or that everything has been said which may be said, yet I do think that we may say within bounds that these two papers together give us a wider range and view of the whole subject than we could get in any other way; that there is no other way, perhaps, in which we could so readily get a satisfactory and comprehensive view of the whole subject of engine balancing. If we go through these two papers and study them carefully and make them as much as possible our own, I feel that we shall have a very thorough and very satisfactory understanding of the problem of engine balancing, its character, the means available for its solution and the various ways in which solutions may be attempted and can be reached. As I said a moment ago, we are not to assume, perhaps, that the last word has been said on this subject; but I think that we may say that we now have the solution well in hand. We know just what the solution requires. We know the various ways in which it may be sought and obtained, and it seems to me the work immediately ahead of the engineer who is seeking to advance this part of the subject will not be so much in the way of seeking new solutions or of further mathematical investigation, but work which shall aim to adopt methods to ready use in the drawing rooms. I have had, on more than one occasion, draughtsmen and others who have been some little time away from their mathematics, and away from those

processes which render them nimble in the use of mathematics, say to me: "This is all very well; I would like to follow out this or that method of investigation; I would like to apply some such method, but it is too much work; these things are not familiar to me; I am rusty on these subjects; is there not some way in which we can simplify the thing?" Of course, these remarks are right along the line of the last speaker. Is there not some way in which this kind of work can be done by someone else once for all, and cannot we simplify the operation so that there shall be some simple, plain road along which I can walk and in which I shall not be liable to make a mistake, and which shall lead me to the results indicated in this investigation? It seems to me that the work of the engineer in the future will be largely along those lines; the adaptation of these elegant mathematical methods, which have been so exhaustively discussed and presented in such a way that the practical man shall have at the minimum of trouble a solution of the problems which arise in his experiments.

MR. CHARLES P. WETHERBEE, *Member*:—The last speaker leaves very little for me to say. I agree most thoroughly with everything he has said. I wish to congratulate Mr. Taylor and Admiral Melville on their papers. I was going to say, if Professor Durand had not spoken otherwise, that the literature is closed on the subject. The method that Mr. Taylor has shown us is really a short cut to the solution of the problem that has, I think, bothered all engineers in designing fast-running engines. The principal difficulty has been in the use of the analytic methods. Take an engine in a certain condition, the results can be worked out; but when it comes to saying what to do to improve the conditions, the previous solutions of the problem have been very unsatisfactory. The analytical methods of determining the effect of the connecting rod, while they have greatly lessened the difficulties by the use of a model rod, and swinging it to determine its moment of inertia, radius of gyration, and elements of that sort; still, when it is all done, you have only determined the forces on the engine, which you have already designed and whose crank angles are already arranged, and it is a foregone conclusion from thereon. Now, this method eliminates all that. I wish to say, also, that, in designing, I have had to do most of the computing myself, and I can appreciate the value of graphical methods. When it comes to the application of these papers, there is always a practical limit as to how close to the ideal you *can* go or should go. I have found that it is not necessary to get absolute balance in an engine, even in the very fastest modern engines in very light vessels, and it is a good deal to know just how far you can go. The three-cylinder engine is not balanced by a good deal, but it can be made by following the methods indicated by Mr. Taylor, sufficiently well balanced for all practical purposes, and it has great advantages in the way of simplicity, etc. Of course, every effort has to be made to get the axis of the cylinders as close together as possible, to have the reciprocating parts of the same weight, as nearly as possible, in all cylinders. Now, as to the single-cylinder

engine Mr. Taylor spoke of, he passed over that very quickly; but the single engine still seems to have a good reason for existence; it still makes a good engine to drive a blower, and for that reason the balance of this engine should be very carefully attended to. It is a question whether we should do as Mr. Taylor indicates this morning, simply balance all the rotating weights by counter-weights (which, of course, leaves a large unbalanced force still in the direction of the axis of the engine), or whether we should overbalance the rotating weight to such an extent that we have got the minimum lack of balance in the two directions by averaging the lack of balance in the line of the axis of the cylinder and across it. I think, perhaps, we should overbalance the rotating weights a little. Now, as to the position of the revolving weights in an engine of more than one cylinder, and the location of the counter-weights to balance the otherwise unbalanced forces in each engine, Mr. Taylor points out that the proper way to do it is to put the weights as far apart as possible, one on each end of the engine, and thus form a couple which is in the proper plane. This solves the problem with the minimum weight; but, in many cases of a fast-running engine, the cylinders can be put closer together than the crank-shaft will allow you to have them, if you have proper bearing surfaces with this method. The addition of counter-weights at each cylinder reduces the pressure on these bearings and enables you to use smaller or shorter bearings and thus contract your engine in a fore and aft direction.

I can only conclude my remarks by stating that the papers, both of them, are of such an exhaustive nature that it is almost impossible to discuss them on such brief acquaintance, and expressing how greatly indebted I am, both to Mr. Taylor and Admiral Melville, for their most instructive papers.

PROFESSOR D. S. JACOBUS, *Member*:—I wish to express my appreciation of the great value of the two papers which have been presented. The subject of engine balancing is indeed an intricate one, and it has been handled by the authors in such a comprehensive way that their work cannot fail to render most useful assistance to those who may make a study of the subject, by placing on record the methods adopted in the best practice, and by presenting the most advanced theories so far developed.

Some time ago I derived formulæ for the exact forces required to accelerate the mass of the connecting rod, and computed certain factors, the use of which would simplify their application. Where the moments required to produce the acceleration of the connecting rod about its centre of gravity are not considered, as in Mr. Taylor's analysis, the formulæ which he gives are all that are required; should, however, these moments be studied, the formulæ and tables which I have prepared may be useful.

In addition to the notation used by Mr. Taylor let:—

kr = the principal radius of gyration of the connecting rod.

X_w, X_c, and Y_w, Y_c = the x and y components of the forces which will accelerate

the connecting rod if applied at the wrist and crank pins, respectively, the x direction being that parallel to the cylinder axis, and the y direction that at right angles to the cylinder axis :—

M = moment required to rotate the rod about its center of gravity.

Then we have

$$X_w = \frac{W}{g}\omega^2 r\left[m\cos\theta + \left[m^2 + \frac{k^2}{n^2}\right]Z\right]$$

$$X_c = \frac{W}{g}\omega^2 r\left[(1-m)\cos\theta + \left[m - m^2 - \frac{k^2}{n^2}\right]Z\right]$$

where

$$Z = \frac{n^2\cos 2\theta + \sin^4\theta}{(n^2 - \sin^2\theta)^{3/2}}$$

$$Y_w = \frac{W}{g}\omega^2 r\left[m - m^2 - \frac{k^2}{n^2}\right]\sin\theta$$

$$Y_c = \frac{W}{g}\omega^2 r\left[(1-m)^2 + \frac{k^2}{n^2}\right]\sin\theta$$

$$M = \frac{W}{g}\omega^2 r^2\frac{k^2}{n^2}\sin\theta\left[\sqrt{n^2 - \sin^2\theta} - Z\right].$$

These formulæ, together with values of Z for several ratios of connecting rod length to crank radius, were published in 1889, in the Transactions of the American Society of Mechanical Engineers.*

In place of the exact value of Z the approximate value of $\frac{\cos 2\theta}{n}$ may be used if so desired.

NAVAL CONSTRUCTOR JOSEPH H. LINNARD, *Member*:—A point which seems to me not to have been touched upon in comparing these papers is one which I understand is really, you may say, the fundamental difference between them. I have not had time to read these papers carefully. If I understand it, Mr. Taylor, in using his graphical methods, and in considering the whole problem, regards as being unimportant the higher period vibrations produced by the obliquity of the connecting rod, etc., which are multiples and higher multiples of the period of the revolution of the units. Admiral Melville, in his paper, lays great stress on these high period vibrational forces and moments, and says, with justice in one sense, that, if they are of equal, absolute value, occurring with a multiple number of frequencies, their total amount is much greater. But I think those of us who have experienced

* Transactions of American Society of Mechanical Engineers, Vol. XI, pages 492 and 1116.

vibrations in a ship find that the vibrational period of a ship rarely goes up to those higher multiples, showing that the ship cannot respond, it seems to me, to the higher vibrational periods upon which so much stress is laid in Admiral Melville's paper. I do not pretend to be an expert in these matters; like some of the others, I am very rusty on my X's, but it is a point that does not seem to have been discussed, and which I would like to see more elaborately elucidated if possible, so that we may know why these differences of opinion exist, and if there are means of reconciling them.

MR. F. L. DUBOSQUE, *Member*:—I have tried to follow this paper through, and what I have to say may be because I have not been able to follow it clearly. But, after reading the article by Mr. Yarrow on balancing marine engines by the use of bob-weights, I was very much impressed with the possibility of correcting a very bad design of engine that we happened to have in a tug-boat, which the pilot claimed almost knocked the pilot-house from over his head, the vibration of the boat was so excessive. We carried out, as nearly as we could, the ideas of Mr. Yarrow in placing weights to counterbalance the moving parts of this engine, and I looked forward with a great deal of pleasure to the trial trip of the boat. After we had gone out and came back the pilot said, "It is worse than before." A little further investigation showed me that I had not got the proper fore-and-aft position of the centre of gravity. Now, in these papers I do not find, if I followed them rightly, that very much attention has been given to the fore-and-aft position of the centre of gravity. I noticed that Mr. Taylor, in his four-cylinder engine, places his low-pressure engine on the end and reduces the weight of the moving parts as much as possible. But it occurred to me that this arrangement was not clearly brought out. Possibly it will be referred to in the response to the discussion.

MR. WILLIAM H. FLETCHER, *Member*:—The next thing to being able to write a prescription is to testify to its results. It is a little peculiar that some of us who were brought up in the home of the beam engine have had experience with both of the methods suggested by these papers,—my firm having built a twin-screw engine, designed by Mr. Macalpine,—a three-cylinder engine,—and installed it in the steam yacht Sovereign, now the Scorpion, United States Navy; and had we not built the Corsair shortly afterwards, we should have been prepared to say there was nothing equal to that engine. We have since built the Corsair, four-cylinder triple-expansion engine, with the cylinders distributed, as recommended by Mr. Taylor; that is, the high and intermediate in the center, two lows at the end, and the air-pumps connected to the low-pressure cylinders, and I think it is only fair to say that, as far as we are able to judge, it is about as near perfection as a man can hope to attain in anything connected with a steam engine. I do not mean to say that balancing cannot go further, but I only mean to say it is only fair to testify to the satisfactory results of the application of the theory as laid down.

THE CHAIRMAN :—In the engine that you refer to, Mr. Fletcher, the cranks were balanced by the Yarrow-Schlick-Tweedy method, were they not?

MR. FLETCHER :—Yes, they were balanced by the Yarrow-Schlick-Tweedy method. We sent them the drawings and the weights of all the moving parts, and they gave us the position of the cranks; we did not pretend to work it out. But I will say this, that we tried to adhere and did adhere in every particular to the weights we sent them, and, so far as the vibration of the engine and the vibration of the boat was concerned, it would seem as though little could be desired. The wheels, I might also say, were very carefully balanced.

MR. WETHERBEE :—This matter of the balancing of the wheels I omitted to mention before. I find, as stated by Mr. Taylor, it is very possible to have a propeller wheel that is perfectly balanced dynamically and still be the cause of great vibration, owing to the fact that the pitch is not the same on all the planes; and, in a propeller which is cast in one piece, this very frequently occurs from the fact that the propeller is bored to an axis which is not the designed axis of the propeller. The propeller comes from the foundry and is put in the boring mill. Ordinarily, the machinist puts the different blades at the same height above the mill on a block, and if there is any error at all in the casting, this results in having the propeller bored in an axis not the designed axis; and, for that reason, one of the blades will be of different pitch—or, rather, they may all three be of different pitch, or we may have two alike and one of different pitch; and I have found that this is a matter which requires a great deal of attention—the placing of the propeller on the boring mill. With a built-up propeller, the same thing applies; when the propeller is prepared to be bolted up, the angle of the blades should also be determined; a way I have found satisfactory is to put the propeller in the boring mill in the same way, clamp a bevel to the tool-post, and set all the blades so that they have the same bevel.

THE CHAIRMAN :—Is there any further discussion? If not, I will ask Naval Constructor Taylor to complete this discussion on his part.

NAVAL CONSTRUCTOR TAYLOR :—Mr. Chairman, I have very little to say in reply to the discussion. I wish to thank all the gentlemen who spoke so kindly of my paper,—much more so than it deserved,—also to extend my commiseration to the Committee. I think that I should say, with reference to some statements made by Mr. McFarland, referring to my paper of 1891, that that paper was by no means so complete as the present, since, although the methods used were the same as in this paper, I carried them only as far as applied to the primary forces; I neglected the effect of the obliquity of the connecting rods. That paper was also on the propeller So far as I know, the first person who made the extension of applying these graphic methods, was Herr Knoeller, Professor in the Technical High School of Vienna.

He published, in 1897, in the transactions of the Austrian Society of Engineers and Architects, a paper on the subject, in which he extended those methods to cover both the connecting rod and the effect of obliquity.

With reference to Professor Peabody's statements, I think you will find that Table 1, on page 62, gives exact values of accelerations of the reciprocating weights for various stated connecting-rod ratios. Exact values are compared with the values from the approximate formula, and this was made for three different connecting-rod ratios 4, 7, and 10, in order to show how rapidly the formula approached the exact value as you increase the length of your connecting-rod, the minimum error for a connecting-rod ratio of 4 is about 6/10 of 1 per cent. of the maximum acceleration, and for a connecting-rod ratio of 10, the maximum is about .046 of 1 per cent. of the maximum acceleration, showing how rapidly these secondary forces fall off.

With reference to the point raised by Mr. Linnard concerning the higher period forces, the fact that the approximate formula agrees so closely with the exact values shows that these higher period forces are small. I think that it is a question about which there seems to be some difference of opinion—not on my part, because I did not express any opinion—the question as to the effect of these higher period forces, and it is one that can only be settled by practical experience, and about that we have very little information at present. I do not believe myself that a structure, such as a ship, as Mr. Linnard said, can respond to forces having a frequency of two or three thousand a minute, which you get up to very rapidly with the fourth and sixth period forces of an engine making several hundred revolutions a minute. That is a question which certainly needs further investigation, but I do not think it can be said to have been established one way or the other. Another reason why I neglected these higher period forces was that my object was to discuss what could be done with various numbers of cranks. As I went on to the second part of my paper, it seemed to be unnecessary to go into this, as we cannot get rid of the higher period forces with less than six or eight cylinders at least. They are very small anyhow. I think if Mr. DuBosque will read my paper carefully, he will find that the question he raised with reference to the fore-and-aft position of the center of gravity was discussed. I have no system to advocate. My only object was to get the facts in the case.

With reference to the Yarrow-Schlick-Tweedy four-crank balance, I do not think they restrict themselves to four cranks. As pointed out in my paper, it is not a complete balance. You will find there is a secondary couple left, and to determine the amount of that couple in any case you will find Diagram G, the last diagram, gives the residual secondary moments. That, taken in connection with the preceding diagram, is for an engine in which the primary forces are balanced and the secondary forces also are balanced. As to the necessity of going to five cranks where you have no forces of a lower period than the fourth power, both primary and secondary forces and moments are readily balanced with five cranks.

That is a matter that is readily settled by practical experience. Mr. Wetherbee

thinks it is not entirely necessary to get rid of primary forces, and Mr. Fletcher seems to think that the balance applied on the Corsair (where there were undoubtedly secondary moments, because the Yarrow-Schlick-Tweedy system cannot get rid of secondary moments) gave results which, from the practical point of view, were all that could be desired. If it is absolutely necessary to get rid of secondary moments it will be necessary to use some such system as that proposed by Mr. Macalpine, or else to go to the five cranks. I do not think it can be done otherwise.

REAR-ADMIRAL GEORGE W. MELVILLE, *Vice-President* (Communicated):—As I was unfortunately not able to be present at the meeting, I avail myself of the courtesy of the Society to contribute a few remarks in comment on the discussion. First of all, I want to express my appreciation of the fact that all of the speakers who referred to the question of *perfect* balance, endorsed the position taken in my paper that the Macalpine system gives this perfect balance. Mr. Taylor remarked that, for perfect balance, either the Macalpine engine, or one having five or more cranks would have to be used. This, of course, means that the Macalpine engine is the best practical solution, for the "multi-crank" engine takes up too much room and is too complicated, while in some ways the Macalpine engine is even simpler than the ordinary four-crank engine.

It seems apparent from the general tone of the discussion that there is an opinion among those who have not studied the subject thoroughly, that an approximate solution of the problem is all that *is* necessary, and that the effort to get a complete solution is an over-refinement. There is, perhaps, the excuse for such a view that, until my paper there was no numerical statement of the amount of the unbalanced forces and moments of higher periods than the first. The analytical discussions had clearly shown their existence, but I realize thoroughly that formulæ do not appeal to busy men, especially when they have only a few minutes for their consideration. It must be manifest, however, that an unbalanced moment of 1,910 foot-tons, as shown in the case of the Deutschland, for second period, is altogether too great to be neglected. I have been informed by intelligent observers who have traveled on the Deutschland, that there are at times very marked vibrations, proving that the lack of balance for higher periods does have an effect, and a very sensible one. Indeed, Herr Schlick admits the vibration, although he attributes it to the propellers. As I have stated in the paper, it seems unreasonable to assume anything but the best design and workmanship in the propellers as in the rest of the machinery. It seems much more reasonable to ascribe the vibrations to an obvious and sufficient cause, viz., the large unbalanced moment—1,910 foot-tons—of second period.

Another point deserves notice, based on a remark of Mr. Fletcher's. He speaks of the performance of the Corsair, and concludes, because there was no vibration, that this is a proof of the excellence of the Y. S. T. system. His remark assumes, of course, that there would have been vibration with a different arrangement of

engine, but we have no reason to anticipate that the revolutions of the engine would necessarily synchronize with the natural period of the hull, for his own statement about the three-cylinder engines of the Sovereign, which we know could not be perfectly balanced, shows that they gave no trouble. Professor Peabody's remark that he knew of engines on the Y. S. T. system which *did* cause vibration is very much more important, because this is definite proof. If all ships were sure to vibrate unless some form of balanced engine were used, Mr. Fletcher's case would be proof of the efficacy of the Y. S. T. system; but as no other form of engine was tried, and we do not know that the engine revolutions would coincide with the natural period of the hull, it proves nothing.

Mr. Linnard's remark about the difficulty of the hull responding to higher period impulses deserves consideration, and Mr. Taylor, although not expressing an opinion, thinks that it has not been demonstrated that the higher period forces do affect the hull. I would call attention to my extract from Mr. Macalpine's paper in the first paragraph on page 99, where he tells of the observations made by Mr. Flood and himself on the Circassia. It is to be noted that these were made in 1892 and were reported about the same time in "Engineering," long before Mr. Macalpine had thought of his present design of engine, so that they cannot be suspected of being the result of a desire to prove the need of his engine. The second paragraph on page 90 which quotes from Mr. Yarrow's torpedo-boat experiments, also gives the strongest confirmatory evidence. He found that there were "*severe* vibrations" at 200, 400, and 800 revolutions, or, in other words, at first, second, and fourth periods. It is true that the engines were running at the various speeds, so that, as far as the engines themselves were concerned, they were first period vibrations, but the point is that the hull responded to each of these sets of impulses with *severe* vibrations, although the relations were of first, second, and fourth period.

Since the meeting, through my attention having been called to the doubt expressed as to the possibility of the hull responding to the higher period impulses, I have gone over the literature of the subject with more care and I find a very striking bit of testimony to the fact that these higher period impulses do show in a very marked way in causing hull vibration. Mr. A. Mallock has studied the vibration problem for many years, and has been employed by the British Admiralty on several occasions to make observations for vibration and report upon them. His remarks, therefore, are those of an expert, and entitled to the greatest weight.

In 1895 in some remarks made before the Institution of Naval Architects, he says, referring to a report made by him in 1892:—"We began with the first order, which is 125 per minute. These are oscillations in the furthest plane; next, 226; next, 450; next, 770; next, 1,420, and next, which was only observed imperfectly, 1,950. All these are vibrations in a vertical plane."

In concluding his remarks, he says:—"The only remaining remark I have to make is with regard to ships, torpedo-boats, and so on. I am sure any naval officer would be willing to stand a considerable amount of slow vibration, but the annoying

thing is the exceedingly rapid vibrations which make movable objects unwilling to stay in their proper places, and prevent writing or reading with any degree of comfort. That going on for days is most trying. The first vibration is only 120 or 130 per minute, and you might have a good deal of that and not mind them very much. *The only way to get rid of these tremors is by having the engine thoroughly balanced*, and I believe any little extra weight that would be involved in doing this would earn its passage over and over again in the comfort and readiness for service which the absence of vibration would bring about. I think anybody who has been in the chart house of some of our ships going at full speed must have experienced a good deal of trouble in keeping charts on the table, and would realize *that the prevention of these rapid vibrations is a most desirable thing.*" (The italics are my own.)

It seems to me, in view of these three cases which I have quoted, there can be no doubt as to the reality of the response of the hull to these impulses of higher periods.

It is easy to understand that a system which deals with a familiar type of engine is more attractive than one which, like the Macalpine system, involves a type of engine which, while composed of well-tried elements, is, as a whole, new. If it were true that the higher period impulses were negligible, and gave rise to no appreciable vibration, the discussion of the Macalpine engine would be purely academic, and I should feel that I was mistaken in my audience, as such a discussion would properly belong in a meeting of mathematical experts rather than an engineering society. I believe, however, that careful study of the analytical treatment of the subject, backed by the admitted cases of vibrating hulls which have Y. S. T. engines, will convince the practical engineer that the Macalpine system, which all admit to give *perfect* balance, is worth most careful consideration. All that is needed is to find an owner who is also posted technically, and Colonel Stevens is such a one, par excellence, and who can appreciate the subject enough to make him willing to give the system a fair trial.

The way in which the subject has been treated in the two papers, and the discussion, has not only had the effect of putting a clear statement of the balancing problem in convenient form, but it shows, I think, quite clearly that the important point remaining for a clear understanding of the subject by engineers who have the time to follow the analytical discussion of the subject, is to present the discussion of the influence of these vibrations, due to high period impulses, in such a way as will be convincing. I regret very much that I did not fully appreciate this aspect of the question when my paper was prepared, but if it is not exploited more fully before the next meeting of the Society, and if my other engagements will permit, I shall hope to present at that time a further discussion of the subject which will, I trust, put this part of the discussion in such a satisfactory form that we can all agree upon it.

THE CHAIRMAN :—Will Mr. McFarland close the debate?

MR. MCFARLAND :—I am reminded of the story of Edward Everett Hale, whose double was told to say that there had been so much said and so well said that nothing remains for me to say except, as we know, there has been already proposed a motion of thanks to the authors of these two papers, and I would like to add a vote of thanks to the gentleman who offered the prize which led to these papers being brought out. I think that the Society is very much to be congratulated on these splendid papers which have been presented and which have made available for us all such valuable information.

THE CHAIRMAN :—Gentlemen, you have heard the motion of Mr. Kafer, supplemented by that of Mr. McFarland, that the thanks of this Society be given first to the authors of these papers and to the author of the scheme by which these papers were brought out, Colonel Stevens. It does not say anything about the Committee—sorry for that.

MR. MCFARLAND :—Please consider that we also want to thank the Committee. That was an oversight.

THE CHAIRMAN :—The Committee will stand upon its dignity and take satisfaction in duty well performed.

(The motion was then put and carried unanimously.)

THE CHAIRMAN :—We have now come to the conclusion of the program for the day, and you will notice that to-morrow has a very interesting list of papers, and I trust that you will all be present, and at the same time that you will not forget the banquet, and that you will make arrangements for being present. The Council will hold a meeting after to-morrow's forenoon session, for the consideration of applications for membership. It is requested that those intending to propose candidates submit the necessary papers without delay.

If there is no further business, gentlemen, the meeting will stand adjourned until ten o'clock to-morrow morning.

SESSION OF FRIDAY FORENOON.

The meeting was called to order at 10.35 A.M., on Friday, November 15th, Colonel E. A. Stevens, Vice-President, in the Chair.

THE CHAIRMAN :—The Society will kindly come to order. The first paper this morning is "A Brief Comparison of Recent Battleship Designs," by Naval Constructor H. G. Gillmor, U. S. N. Naval Constructor Taylor will kindly read the paper in the unavoidable absence of the author.

Mr. Taylor read the paper.

A BRIEF COMPARISON OF RECENT BATTLESHIP DESIGNS.

By Naval Constructor H. G. Gillmor, U. S. N., Member.

[Read at the ninth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 14 and 15, 1901.]

There is probably no class of design work in which what is to be done is so largely controlled by what has been done, and what is being done, as in the designing of naval vessels. The value of any vessel for naval purposes must necessarily be determined finally not only by the features embodied in the vessel itself, but by the characteristics of the vessels against which she may be opposed. It is this which makes the study of the development of foreign designs a necessity, and determines the characteristics of the vessels of the several classes which may at any time be in course of design.

While from time to time new types of vessels have been introduced and developed, there has always been a type of vessel recognized in each period as the main strength and backbone of naval force; and this type has always been designated the battleship. It is to the consideration of the designs of vessels of this class of the present period that attention is asked. For the purposes of comparison there have been chosen the most recent designs of the several principal naval powers; and the comparison will be confined to the design conditions for those features which directly contribute to the naval value of each unit—that is to say, the armament, the protection, the speed, and the coal supply. The designs chosen are Great Britain, the Duncan; Germany, the Wittelsbach; Russia, the Borodino; Italy, the Vittorio Emanuele; Japan, the Mikasa, and the United States, the Virginia. In each case there are represented two or more vessels, the naval features of which are virtually the same as that of the vessel named. The information available with respect to the latest French battleship design is so meagre as to have made it impracticable to include it in the comparison proposed.

In Plate 47 small scale sketches, showing these vessels in elevation and deck plan, are given with the purpose of presenting visually the differences in naval features among them. Full data is given in the tabular statement in Table I which follows.

TABLE I.

	Virginia.	Duncan.	Borodino.	Mikasa.	Wittelsbach.	Vittorio Emanuele.	Type design.
Laid down	1901	1899-00	1900	1899	1899-00	—	—
Length between perpendiculars	435′	405′	397′	400′	418′	435′	435′
Breadth, moulded	76′ 10″	75′ 6″	76′ 0″	76′ 0″	68′ 0″	73′ 6″	76′ 0″
Draught, mean	24′ 0″	26′ 6″	26′ 0″	27′ 2″	25′ 0″	25′ 7″	—
Displacement, in tons	14,950	14,000	13,600	15,200	11,800	12,624	—
Indicated H. P. with forced draught	19,000	18,000	16,000	15,000	15,000	19,000	—
Speed, in knots, with forced draught	19.0	19.0	18.0	18.0	19.0	22.0	18.0
Boilers	Various.	Bellville.	Bellville.	Bellville.	Cyl. & Schultz.	—	—
Armament:							
Main battery	4-12″	4-12″	4-12″	4-10″	4-9.4″	2-12″	4-9.4″
	8-8″	12-6″	12-6″	14-6″	18-6″	12-8″	12-6″
	12-6″	—	—	R. F.	R. F.	—	—
Secondary battery	12-3″	12-12 pdrs.	20-12 pdrs.	20-12 pdrs.	12-3.5″	12-3″	12-12 pdrs.
	8-3 pdrs.	6-3 pdrs.	20-3 pdrs.	8-3 pdrs.	12-1 pdrs.	12-1.8″	6-3 pdrs.
	8-1 pdrs.	—	4 Maxim.	4-2½″	—	—	—
Torpedo tubes:							
Above water	—	—	4	—	1	4	—
Submerged	2	4	2	4	5	—	2

TABLE I—(Continued).

	Virginia.	Duncan.	Borodino.	Mikasa.	Wittelsbach.	Vittorio Emanuele.	Type design.
Protective deck:							
Thickness of slopes.....	3″	1″	2½″	2″	3″	4″	3″
Thickness of horizontal parts.....	1½″	2″	1½″	—	1½″	2″	1½″
Armor:							
Length of water-line belt.	Whole length.	⅔ length.	Whole length.	Whole length.	Whole length.	Whole length.	⅔ length.
Breadth of water-line belt.	8′ 0″	7′ 0″	6′ 6″	7′ 9″	7′ 0″	—	7′ 0″
Thickness of water-line belt at amidships	11.8″	7″	8″	9″	8.8″	9¾″	7″
Thickness of water-line belt at ends.....	4″	2″	5¾″	4″	4″	4″	2″
Bulkheads.....	6″	7″	—	6″	6″	6″	6″
Length of upper belt.....	⅔ length.	⅔ length.	Whole length.	⅔ length.	¼ length.	⅓ length.	¼ length.
Width of upper belt.	—	7′ 0″	5′ 6″	7′ 6″	7′ 6″	7′ 6″	7′ 6″
Thickness of upper belt..	6″	7″	6.4″	6″	6″	6″	6″
Protection, largest guns.	10″	11″	10″	14″	10″	8″	8″
Protection, medium calibre guns.....	6″	6″	6″	6″	6″	6″	6″
Protection, other guns....	2″	—	3″	—	—	—	—
Conning tower, forward.	9″	12″	—	14″	10″	10″	9″
Conning tower, after.....	5″	3″	—	3″	6″	—	3″
Coal supply, normal.....	900	900	900	1,400	650	1,000	650
Coal supply, bunkers full....	1,900	2,000	1,500	2,000	1,250	2,000	1,250

Even to the casual observer considerable variation in the design features must be noticeable; and since this variation may be taken as an indication of the range within which designers of the present have considered it advisable to limit these elements, a brief notice of them may not be without value.

In armament, uniformity of practice is found in the location of the heavy guns, but in their number and calibre a considerable range is noticeable. With but two exceptions 12-inch guns have been chosen, and, with a single exception, the number of heavy guns is four, mounted in pairs, two forward and two aft. In the Mikasa, four 10-inch guns make up the heavy battery, though in other vessels of practically the same general characteristics built for the Japanese government, 12-inch guns have been employed. In the Wittelsbach design, four 9.4-inch guns have been adopted, marking the limit in lightness of calibre of the first calibre guns in battleships. The one exception to the number four for the heavy guns is in the Vittorio Emanuele, where two 12-inch guns, mounted one forward and one aft, have been chosen. In the guns of second calibre an almost equal range in calibre and a considerable variation in arrangement are noticeable. In all but two of the designs under consideration what may be called a central battery, more or less extended, is employed. This central battery reaches the extreme of concentration on the Wittelsbach, and is most extended on the Virginia. In two out of the six designs the secondary calibre guns are mounted exclusively in such a central battery, with the addition of four guns of second calibre in isolated casemates, giving bow and stern fire. In two others, in addition to the central battery, a portion of the second calibre guns are mounted in turrets, giving fire over a considerable arc from directly ahead or directly astern. In these turrets there are, in one case, guns mounted in pairs, and, in the other, single guns. The Borodino and Vittorio Emanuele mount the second calibre guns entirely in such turrets. In the Wittelsbach, the central battery, isolated casemates and turrets, for the second calibre guns, are used in combination. In the choice of a calibre for second calibre guns, 8-inch guns and 6-inch guns divide honors. In the Vittorio Emanuele the second calibre guns are 8-inch guns. In all the others, with the exception of the Virginia, the second calibre guns are 6-inch guns. In the Virginia both 8-inch and 6-inch guns are employed. The number of guns of second calibre varies from twelve 6-inch in the Duncan to twelve 6-inch and eight 8-inch in the Virginia. In the secondary battery also there is considerable variation in the number and calibre of the guns, the arrangement being dependent upon

the system employed in mounting the second calibre guns. The calibre ranges from 15-pounders and 1-pounders in the Wittelsbach to 12-pounders and 3-pounders in the Duncan; and, in numbers, from forty in the Borodino to eighteen in the Duncan.

In the matter of protection to stability by a water-line belt, a considerable range is noticeable. The lower limit is found in the Duncan, in which two-thirds of the length of the vessel is protected by a belt of a thickness of 7 inches, the forward end being protected from the fire of secondary battery guns by 2-inch nickel steel plating riveted upon the skin plating—as in protective decks. The remaining five vessels under consideration have, for stability protection, complete water-line belts, the maximum thickness of which is found in the Virginia—11 inches at top of armor amidships, tapering to 8 inches at the bottom. The lower limit in the extent of the upper belt, affording protection to the ammunition supply for the battery, is found in the Wittelsbach, where this belt is limited to about one-fourth of the length of the vessel. Three of the vessels under consideration, namely, the Virginia, the Duncan, and the Mikasa, carry this upper belt over about two-thirds of the length. The Vittorio Emanuele has an upper belt over somewhat more than one-third its length, and the Borodino for the whole length. It should be mentioned, however, that in the Borodino both the water-line belt and this upper belt are so narrow that the two combined really make a wide water-line belt over the whole length of the vessel. The thickness of the armor employed for this upper belt is uniformly 6 inches, except in the case of the Duncan, in which the water-line belt and upper belt are continuous and 7-inch armor is employed.

In the character and extent of the protection to the guns of first calibre, the greatest range is noticeable. The lower limit of such protection is found in the Vittorio Emanuele, where 8-inch armor is employed, and restricted armored tubes with walls of equal thickness, for protection to the ammunition supply. From this the degree of protection ranges upward, as may be readily observed from the figures in Plate 1, the maximum being found in the Mikasa, in which there are barbettes 14 inches in thickness extending at their full diameter to the top of the water-line belt.

The protection adopted for the guns of second calibre is uniformly 6-inch armor throughout all the designs under consideration. The method of disposing this armor for the protection of the guns varies. In three of the six ships there are central batteries, concentrated in the Wittelsbach and extended in the Virginia and Mikasa. In addition to this central battery the Wittelsbach and Mikasa employ, also, isolated casemates with 6-inch armor

for a portion of the guns of second calibre; and, in the Duncan, this system of protection exclusively is employed for the second calibre guns. Upon the Vittorio Emanuele and Borodino, the second calibre guns are all carried in armored turrets, with 6-inch armor, and, in the Wittelsbach and Virginia, such turrets are combined with central batteries, or with central batteries and casemates. The exceptional features in protection and system of mounting in the superposition of 8-inch turrets upon 12-inch turrets in the Virginia, has been the subject of so much discussion as to make special comment unnecessary. The Borodino and Virginia are the only vessels in which any protection is provided for guns of the secondary battery.

The speed adopted for two of the vessels is 18 knots. In the Virginia, Duncan, and Wittelsbach, the designed speed is 19 knots. In the Vittorio Emanuele the speed feature is developed to an extent which separates this vessel widely from the others under consideration. Her speed of 22 knots, if realizable, would practically place her on an equal footing, as to speed, with the armored cruisers being constructed by several of the naval powers. There seems reason to doubt the accuracy of report as to the speed feature of this vessel, regard being given to the other features proposed.

The lower limit in the coal supply in the designed condition is found in the Wittelsbach, the design of which provides for a normal supply of 650 tons, with a maximum stowage capacity of 1,250 tons. From this the normal coal supply ranges through 900 tons for the Borodino, Virginia, and Duncan, 1,000 tons for Vittorio Emanuele, to a maximum for this element of 1,400 tons in the Mikasa.

The variations which have been pointed out make direct comparison of the several vessels difficult. To establish a basis for the present comparison, it is proposed to assume a vessel whose dimensions are those of the largest vessel under consideration, in which the features of armament, protection, speed, and coal supply embodied are the minima of these several features, which may be found among the designs under discussion. Such a vessel would represent the extreme limit to which, as judged from current practice, it is thought possible to reduce the several elements, and will be designated a *type design*.

This *type design* will then be a vessel of about 435 feet in length and 76 feet in breadth (about the extreme dimensions of the Virginia class), having a speed of 18 knots (the speed of the Mikasa and Borodino); with a normal coal supply of 650 tons (that of the Wittelsbach); a battery of four 9.4-inch guns (the first calibre battery of the Wittelsbach), and twelve 6-inch

guns, twelve 12-pounder and six 3-pounder guns (the second calibre guns and secondary battery of the Duncan; two torpedo tubes (as in the Virginia); a water-line belt 7 inches thick and 7 feet wide, extending from the after barbette for two-thirds of the length of the vessel, with 2-inch nickel steel plating carried from the forward end of this belt to the stem (the water-line protection of the Duncan); surmounted by a shorter belt 6 inches in thickness, enclosed by bulkheads at its ends of the same thickness, extending up to the top of the gun deck to form a protected battery (as on the Wittelsbach); the 9.4-inch guns and ammunition supply protected by barbettes and ammunition tubes, carrying armor 8 inches in thickness (the protection of the first calibre guns of the Vittorio Emanuele); and the 6-inch guns and ammunition supply protected by 6-inch armor (the protection of the second calibre guns provided in all of the designs under consideration). It has been necessary to assume that the protective decks in the several designs under consideration are of practically equal value as protection, and that the type design has an equivalent protective deck. A sketch outline of the elevation and deck plan of this type design will be found at the top of Plate 47, and detailed data relative to her several features in the last column of the foregoing table, Table I.

To attempt to express in figures an absolute or relative naval value, even for vessels of the same class, is generally regarded as almost impracticable, because of the impossibility of suitably assigning values among the several design elements, all contributing to a successful whole and differing so widely in their individual purposes as to be practically incomparable. Since, however, such an expression of value, even if only very approximately correct, affords a means of giving point to a comparison, an effort will be made to reduce the present comparison to such terms as will permit the assignment of approximate relative values. The results, depending as they do upon so many things which, at best, can be but inaccurately known to any but the designer of each vessel, must, of course, be regarded as qualified by the inaccuracies in the data upon which such results depend.

It may fairly be assumed that the naval value of any vessel at a given period depends chiefly upon the battery carried; the protection given to stability, armament, ammunition supply and personnel; the speed; and the time during which she may operate without interruption, as measured by her coal supply. Her naval value, therefore, is independent of her displacement, although there is a relation between that naval value and displacement which fixes the limit of naval value which may be reached upon any displacement, and the excellence of any design should be judged by the nearness of the approach of that design to this limiting relation.

Such a vessel as the type design outlined above has, for naval purposes, a definite value, which might be expressed in a variety of ways. Each of the several vessels whose designs it is proposed to compare may be regarded as being such a vessel as the type design, upon which improvements of value have been introduced in one or more of the elements of armament, protection, speed, or coal supply. The designers of the several vessels under consideration, having provided for the minima of the several essential features of battleship design, have varied the distribution of the remaining disposable weight in a manner which each individually deemed most efficient. Since each of the several features may be found developed to an extreme in some one or more of the vessels being compared, and since the one restricting and governing condition in such development is weight, may it not fairly be assumed that the naval value, assignable to the excessive development of any one of the design features, may be represented by the weight necessary to such a development in any degree above that represented by a minimum embodied in all the designs? Such an assumption will here be made; and the naval value of the type design will be assumed to be 4,300, the approximate weight required, under the conditions outlined, to provide in the type design the features embodied. The relative naval value of each vessel under consideration, as compared with the type design, will be expressed finally by adding to 4,300, the naval value of the type design, the weight which it would be necessary to add to the type design in order to provide in that vessel for the armament, armor, speed, and coal carried in the design whose relative naval value it is desired to represent.

The several vessels, when compared with the type design, show increases in the several naval features, with corresponding added naval value as follows:—

VIRGINIA.

Four 12-inch guns instead of four 9.4-inch guns	Value	160
Eight 8-inch guns additional	"	350
Twelve 14-pounders instead of twelve 12-pounders, and two 3-pounders and eight 1-pounders additional	"	40
Heavier and more extended water-line protection (11-inch armor as compared with 7-inch armor amidship)	"	410
More extended upper belt	"	120
Heavy guns in turrets protected by 10-inch armor; large diameter barbettes with 10-inch armor	"	620
Protection of 8-inch gun positions, and extended central battery protection	"	240
Protection to portion of secondary battery	"	40
Additional normal coal supply	"	250

One knot more speed, which, with weights added above, necessitates increase in machinery weights	Value	520
Total additional value		2750
Value of type design		4300
Relative naval value		7050

DUNCAN.

Four 12-inch guns instead of four 9.4-inch guns	Value	160
Two additional submerged torpedo tubes	"	20
More extended upper belt of greater thickness	"	350
Heavy guns mounted on turn-tables with barbettes protected by 11-inch armor	"	500
All second calibre guns in isolated armored casemates, affording additional protection to such guns	"	260
Additional normal coal supply	"	250
One knot more speed, which, with the weights added above, necessitates increase in machinery weights	"	450
Total additional value		2000
Value of type design		4300
Relative naval value		6300

BORODINO.

Four 12-inch guns instead of four 9.4-inch guns	Value	160
Eight additional 12-pounders and fourteen additional 3-pounders	"	50
Four additional above water torpedo-tubes	"	30
Heavier and more extended water-line protection (9-inch armor as compared with 7-inch armor amidship)	"	100
More extended upper belt, complete	"	200
Heavy guns in turrets protected by 10-inch armor with large armored tubes	"	450
Protection to portion of secondary battery	"	120
Additional normal coal supply	"	250
With same speed, weights added necessitate increase in machinery weights	"	130
Total additional value	"	1490
Value of type design		4300
Relative naval value		5790

MIKASA.

Four 10-inch guns instead of four 9.4-inch guns	Value	50
Two additional 6-inch guns	"	50
Eight additional 12-pounders, two additional 3-pounders, and four additional 2½-pounders	"	20
Two additional submerged torpedo-tubes	"	20

Heavier and more extended water-line protection (9-inch armor as compared with 7-inch armor amidship)	Value	350
More extended upper belt	"	190
Heavy guns mounted on protected turn-tables, with large diameter barbettes protected by 14-inch armor	"	580
More extended central battery and protection to two additional 6-inch guns	"	190
Additional normal coal supply	"	750
With same speed, weights added above necessitate increase in machinery weights.	"	230
Total additional value		2430
Value of type design		4300
Relative naval value		6730

WITTELSBACH.

Six additional 6-inch guns	Value	145
15-pounders instead of 12-pounders	"	10
Additional torpedo outfit	"	25
Heavier and more extended water-line protection (8.8-inch armor as compared with 7-inch armor amidship)	"	130
Additional protection to heavy guns (10-inch armor as compared with 8-inch armor)	"	80
Armored casemate for two additional 6-inch guns and turrets for four additional 6-inch guns	"	240
One knot more speed, which, with weights added above, necessitate increase in machinery weights	"	330
Total additional value		960
Value of type design		4300
Relative naval value		5260

VITTORIO EMANUELE.

Two 12-inch guns instead of four 9.4-inch guns	Value	50
Twelve 8-inch guns instead of twelve 6-inch guns	"	220
Twelve 14-pounders instead of twelve 12-pounders, and six additional 3-pounders	"	20
Additional torpedo outfit	"	30
Heavier and more extended water-line protection ($9\frac{3}{4}$-inch armor as compared with 7-inch armor amidship)	"	270
More extended upper belt	"	80
Protection to second-calibre guns by turrets	"	160
Additional normal coal supply	"	350
Four knots more speed, which, with weights added above, necessitate increase in machinery weights	"	1500
Total additional value		2260
Value of type design		4300
Relative naval value		6560

In Table II, the several vessels will be found in the order of their relative values, and the values assigned to each of the several items are given in such a manner as to facilitate direct comparison of the distribution of weight among the several features.

TABLE II.

TABLE OF RELATIVE, ADDITIONAL, AND TOTAL NAVAL VALUES.

	VIRGINIA.		MIKASA.		VITTORIO EMANUELE.		DUNCAN.		BORODINO.		WITTELSBACH.	
	Added value.	Totals.	Added value.	Totals.	Added value.	Totals.	Added value.	Totals.	Added value.	Totals.	Added value.	Totals.
Heavy guns and ammunition.....	160		50		—50		160		160		—	
Second calibre guns and ammunition.....	350		50		220		—		—		145	
Secondary battery guns and ammunition.....	40		20		20		—		50		10	
Torpedo outfit.....	—		20		30		30		30		25	
Total amount.....		550		140		220		190		240		180
Water-line protection.....	410		350		270		—		100		130	
Upper belt and bulkheads.....	120		190		80		350		200		—	
Protection to heavy guns.....	620		580		—		500		450		80	
Protection to second calibre guns.	240		190		—160		260		—		240	
Protection to secondary battery...	40		—		—		—		120		—	
Total protection.....		1,430		1,310		190		1,110		870		450
Coal supply.....	—	250	—	750	—	350	—	250	—	250	—	—
Speed.....	—	520	—	230	—	1,500	—	450	—	130	—	330
Total additional value.....		2,750		2,430		2,260		2,000		1,490		960
Value of type design.....	—	4,300	—	4,300	—	4,300	—	4,300	—	4,300	—	4,300
Relative naval value.....		7,050		6,730		6,560		6,300		5,790		5,260
Designed displacement.....	—	14,950	—	15,200	—	12,624	—	14,000	—	13,600	—	11,800
Efficiency of design.....	—	47.2	—	44.3	—	52.0	—	45.0	—	42.7	—	44.6

It should be noticed that there is reason to believe that the figures for the indicated horse-power and displacement for the Italian vessel, Vittorio Emanuele, are inaccurate, since, in the present state of shipbuilding art, it would seem to be impracticable to procure, with the horse-power given, the designed speed of 22 knots in a vessel of the designed displacement of 12,624 tons; and this displacement would seem to be inconsistent with the features which, if report relative to this vessel be true, it is proposed to embody in the design. It should also be noticed that the feature of protection given by the protective decks of the several vessels is regarded as being of equal efficiency and value in all the vessels. It is known that in the Borodino this protective feature has been developed to an extent somewhat in excess of that found in the other vessels compared; and, had it been practicable to extend the comparison to include this feature, the Borodino would, doubtlessly, have shown to better advantage in the final results.

Since the relative naval values given above are expressed in terms of, mathematically, the same dimensions as those employed in the expression of displacement,—if the relative naval values be divided by the designed displacements of the several vessels, the results may be expressed as a percentage, which might be termed the efficiencies of the several designs. These figures are given in the last line of Table II, and varying, as will be seen, from 42.7 per cent. for the Borodino, up to 52 per cent. in the Vittorio Emanuele (assuming the designed displacement given for this vessel to be a possible one); and the order of merit of the designs would be, on this basis, as follows:

Vittorio Emanuele, 52.0.
Virginia, 47.2.
Duncan, 45.0.
Wittelsbach, 44.6.
Mikasa, 44.3.
Borodino, 42.7.

In conclusion, it may be stated that while reasonable care has been given to the estimates upon which they are based, detailed accuracy in the figures given above is not claimed. The purpose has been to roughly estimate and express in concrete terms the relative naval values of the several vessels whose designs were considered, in order that the results of this comparison might be presented in a form more tangible than that of a general discussion of the several features of the designs.

DISCUSSION.

THE CHAIRMAN (Mr. McFarland):—You have heard the reading of the paper, gentlemen, and I hope you will give it a full discussion, and I will first call on the head of our own Construction Department, Admiral Bowles. I know that you will all agree with me that it gives us great pleasure to welcome him here with his new title. This is his first appearance since his promotion, and I am sure you will be very glad indeed to hear from the Admiral. (Applause.)

REAR-ADMIRAL BOWLES, *Vice-President*:—Mr. Chairman and gentlemen: I am sorry that the Chairman has deprived me of the privilege I hoped to have of being the last to speak, a situation that has some advantages. This paper presents the subject of battleship design from a remarkably simple and interesting point of view; it is purely the naval architect's point of view. It does not involve any assignment of relative value to the different points in a design, and it is free from all those confusing things called "muzzle energy," "rates of fire," and different things which puzzle people so much who are obliged to consider these designs. But it is simply a question of balancing up tons in which the tons of the different countries are given the same weight. I notice that Russian tons and American tons and English tons are all considered of equal value and efficiency in making up the results and the design. There is one peculiarity of all these methods of evaluating paper designs—I notice that wherever they originate they bring out the local battleships to be the best—a very proper thing, of course. (Laughter.) I think the most serious error in this comparison lies in the neglect of the difference in the protective decks, and the author has probably found it impossible to get sufficient data on that point to introduce it into his comparison, and that leads me to point out one of the differences between our own battleships and some of the foreign designs. The Duncan class, representing all the later English battleships, uses much more weight in the protective deck than any of our vessels, because they carry really a double protective deck, one which covers the top of the main armor belt and involves the use of a large amount of weight which does not appear in this comparison. That would probably affect the relative value of these designs very considerably. That point is the only one which I wish to bring out. But there is a tendency shown by these designs which we may as well face now. When the relative value is given to the different elements in these designs, it will be found that some of the foreign types possess considerable advantages in the individual protection of the battery given by armor, and that is apparently an increasing necessity due to the probable use of high

explosives and the great and increasing power of the modern gun. The inevitable result will be that the battleships will increase in size, and it is to be hoped that while they increase in size they will not increase in complexity. Therefore, you must look for a very considerable increase in size in the battleships to be built for the United States Navy, and I would like to assure you that whether they are built from the designs of the majority or the minority of the Board on Construction, we hope that the supremacy of the individual battleships of the United States Navy will be unquestioned. (Applause.)

THE CHAIRMAN :—I want to say to the Admiral that we hope he will not feel limited by the fact that he has spoken once. We are entirely willing that he should have a chance to speak again when the others have spoken. I believed you would all welcome the chance to show your appreciation of his recent promotion, and I could not resist the temptation to give you the opportunity of expressing your feelings.

We would be glad to hear from one of our members from over the sea, Captain von Rebeur, if he wishes to say anything on the subject.

COMMANDER VON REBEUR, I. G. N., *Associate* :—Mr. Chairman, it is very kind of you, but since I have been for nearly five years in the United States I think that I know probably more about United States battleships than those of recent German design. (Laughter.) It is really difficult to keep up with both, and, of course, comparison between your battleships and ours, while being highly interesting, necessitates having data of my own ships to go into the subject satisfactorily. What always interested me in the battleships of American design is especially the enormous amount of coal which they carry, in which the German battleships are probably behind a little as well as all the other foreign ships; this appears, of course, a rather startling fact, but is based, I believe, on reasons which have been very clearly shown; that is to say, it is largely on account of the coal which the American battleship needs to carry before it can get to another station where it can fill up with coal either by means of colliers or coaling stations. When Admiral Bowles spoke of the constantly increasing size of battleships, I was rather surprised, since it was my belief that you formerly tried to keep down the size of battleships so as to be able at all times to enter your own harbors. I recall this fact because I think it is stated in the last annual report of the Chief of the Bureau of Construction, as published by the papers, which said that on the Pacific coast there is at present only one dock in which your battleships can enter, that is at Puget Sound, since the new dock built at Mare Island Navy Yard cannot at present be approached by battleships of any size, because there has to be extensive dredging. I should be very glad to know how far the increasing size of battleships necessitates deeper draught. Probably we shall learn something about this fact.

THE CHAIRMAN:—We would like also to hear from another of our foreign colleagues, Captain Tchernegowski, of the Imperial Russian Navy.

CAPTAIN TCHERNEGOWSKI, I. R. N.:—I think the members of the Institution will excuse me, but I have just received this paper, and in order to give any opinion of value one would need to look into the paper more deeply. I have not had any opportunity to look at it; so I am not prepared to give any opinion of the qualities of the various ships discussed. I only know that there was proposed a new method to value ships; but all these methods are very good in one respect and very bad in another. In one respect they give good results and in another case quite different. But what I think is generally forgotten is the most important part in the evaluation of a ship, and that is the personnel—the men who manage the ship. You can have a very speedy ship, but if the men who manage it are not capable she will not do credit to her designers; so that everybody connected with the ship must be taken into account, and the technical condition of the country itself must also be taken into account—that is, how the men in the country are trained. Armor protection of different kinds may be of very high value, but its disposition and general arrangement have to be taken especially into account before you can ascribe anything like accurate values.

With regard to the Russian type ship here given, the Borodino, which has nearly the least value of all the ships so far as this comparison goes, I may say that our ships generally require more weight in scantlings and fittings, and there may be appurtenances which others regard as minor affairs, but which require weight and which we consider very important in the management of the ship, and which add greatly to the total result in fixing the real value of the ship and without which we consider the ship is not complete. It is, therefore, difficult to try to compare one ship with another. I think at present I cannot say anything more about this; but, as Admiral Bowles says, you always think your own ship is the best.

THE CHAIRMAN: Captain, we are very much indebted to you for your remarks, and I am sure that we had a very striking illustration in our recent naval experience in actual battle, which showed that the point you made about the technical condition of the country and the importance of the personnel cannot be overestimated. Of course Mr. Gillmor, in his paper, was taking up the ship itself without regard to the personnel, and assuming that every ship had an equally efficient personnel; but that is a very good point to make, and one which we all appreciate, I am sure. Now, we have heard from several gentlemen who have had to do with the building of the ships, and we have one or two here who, in time of war, would have to do with handling and fighting them, and we would like to hear from our old friend Lieutenant Niblack, if he cares to say anything.

LIEUTENANT NIBLACK, *Associate*:—I do not care to make any comment at present. Thank you very much.

THE CHAIRMAN:—We have another gentleman here who is an ordnance sharp and one of the bright young officers of the Navy. Please note the stress on the word "young." We should be very glad indeed to hear from Lieutenant Chandler.

LIEUTENANT CHANDLER, *Associate*:—There is only one thing that struck me in this comparison of weights, to which Admiral Bowles referred. Of course, this method of comparison seems very valuable, and undoubtedly is—the question of weight. I do not know of any more important one, but, at the same time, in comparing efficiencies of ships of different types, I do not see how any satisfactory conclusion can be reached that does not include what Admiral Bowles called the vexing questions of "arcs of fire," and "muzzle energy," and one thing and another of that sort, and I think, to take an extreme case, the necessity for considering that point has been shown in several instances in several countries. I think, notably, if I remember correctly, in the little trouble that the French had with the Chinese, where the Chinese had a number of gunboats which had one gun pointing forward with very little train; and, when they did not get springs on their cables and an ebb-tide set in, those guns were not very valuable against a fleet coming up the river. I think under these circumstances the military value of those gunboats could not be called very great, and to extend that idea further a case might be found where, under this method of comparison, a ship might show very highly, and yet be at a decided disadvantage when fighting, under certain conditions, a vessel which, by the same method of comparison, seemed to be a very much less efficient ship; therefore, I do not think that the result would bear out a paper comparison beforehand. I think that, for a really efficient comparison of ships, these troublesome things have got to come in, and yet, after more or less thought, I do not quite see how it is going to be done. There was a very extensive paper published some time ago, written, I think, by a naval constructor in the German navy, which assigned weights, etc. I started in on that very faithfully, and I got about half through when I concluded that the weights were too weighty for me and had to stop.

LIEUTENANT COMMANDER WHITE, *Associate*:—I am of the opinion that the comparisons made in this paper are based on the obvious only. The sketches of exteriors of ships, even though armor dimensions are given, do not necessarily give a basis for an accurate estimate of the relative protection afforded in the gun positions to the mechanism of the guns or to their crews. If so used, it must be granted that the inferences which may be drawn from these sketches can be based upon our knowledge of vessels already built upon similar lines. The comparisons made in the paper grant equal protection in all cases for the same thickness of armor with-

out regard to disposition or arrangement. If we take, for instance, the protection given the 6-inch guns and their crews, comparing the Virginia with the Duncan, being the extremes of examples of all the guns being in one compartment or open in one deck on one hand, and as separated in closed casemates on the other, this is decidedly not just. For it is possible, if we are to credit the reports of the explosive effect of high explosives, for one large, or even medium, caliber shell bursting in the gun deck of the Virginia, to "knock out" the personnel of the entire battery in this deck. It is apparent that it will take six shots to produce the like effect on the Duncan, while the target exposed in broadside by combining the casemates is the smaller.

In the question of ammunition supply, the individual gun positions meet the requirements of rapid fire, as each gun has its separate line rising from below in the immediate vicinity of the gun. A smaller number of electrically driven hoists may provide ample ammunition for the guns in the open deck, but any derangement of an individual hoist will impair the service of the guns near it, and recourse to neighboring hoists will interfere with the service of other guns.

The comparison does not include all the protection that may be given the gun positions, as it omits the possible gun shields. Certainly the fair sides of the Virginia will require a larger port area for the same angle of train than the Duncan with sponsons. If a gun shield is used in the Virginia, the protection it may give, using the same weight of armor, will not be as good as in the Duncan. Vessels of this class have an opening above the shield of but 3 inches, and it is a question if even this be necessary. This exposure of open port is not to be belittled, if we consider the volume of small caliber fire a ship may deliver and the injury to the mechanism of the gun a small shell is capable of doing. The exposure is even of greater moment in the case of the turret guns. The vertical port of the 12-inch turret of the Virginia must necessarily be greater in area than that of the Duncan, where the inclined shield permits the trunnions of the guns to be brought close to the port, and where the actual opening is remarkably small and as a target inconsiderable.

Comparing the heavy gun positions, the most striking feature in these two vessels is the 12-inch turret of the Virginia with its superstructure. This feature is even more prominent when the vessels themselves are seen. In the far East, I had the opportunity of seeing the Kentucky with other battleships. At a great distance this feature is discernible, discovering the character and nationality of the ship at once, and it would need no telescope to locate the target. The turrets, or rather gun shields, of the British battleships are so low as to be inconspicuous even at a short distance, and owing to uniformity of outline in the designs, the battleship does not differ noticeably from the armored cruiser. Apart from being more conspicuous, the turret of the Virginia gives a target area about three times that of the Duncan, the most dangerous feature of which is the greater height, since errors of elevation are more probable than those of lateral direction. The 8-inch turret makes a most effective save-all, for it increases the target area by one-third, and *its height two-thirds.*

The great weight of the Virginia's turret is decidedly a disadvantage when the possibility of a breakdown to the turret turning mechanism is considered, and recourse is had to hand power.

While the 8-inch guns are more effective at great ranges, yet when we consider the difficulty of keeping the gun pointed at all times at the target, which must be done to ensure rapidity of fire, their slower rate of fire and the small percentage of hits developed at target practice and in action, as compared to what may be done with the 6-inch gun, I think we are hardly warranted in giving them such comparatively greater weight in this comparison, or of handicapping our ships with the extra weights involved by using such a variety of calibers for the secondary work.

THE CHAIRMAN:—We would be very glad to hear from any other gentleman. This is a most important subject, and the discussion is very interesting, and has brought out some points that may be very useful.

ADMIRAL BOWLES:—Mr. Chairman, I would say one word in regard to the subject which was spoken of by one of the speakers, and that was in regard to the draught. The limitation of the draught of vessels of the United States Navy still continues to exercise an adverse influence upon the efficiency of our designs, but we must consider, other things being equal, that, the less the draught of the vessel the greater her usefulness. So that our limitations bear fruit in that respect, and, when we say that the vessels are liable to increase in size, we have got to face the necessity for some increase in draught on account of the great value of the defense which is given by the most efficient battleships. Therefore, I have come to look at the matter somewhat from the point of view of an eminent civil engineer, who remarked to me that, while the Lord made harbors man could remake them.

LIEUTENANT COMMANDER WHITE:—The commercial requirements for increased draught of merchant vessels is leading to the improvement of our harbors and waterways, so that restrictions in draught are not so important as formerly.

MR. G. C. COOK, *Member*:—I see there is no reference made to torpedoes or the efficiency of torpedoes in these designs. I believe most of the battleships are fitted with them, and I would ask if there can be any information given on that subject.

LIEUTENANT CHANDLER:—As far as the putting of the torpedo on board large ships is concerned, I do not think either in this country or any other there is much tendency in that direction. In the United States Navy, we are leaving them out; while in some navies they are using the submarine tube, I do not know

whether this practice is going to continue or not; I have serious doubts. So I think that the torpedo can be pretty well left out of account in battleship construction.

THE CHAIRMAN:—The discussion on this subject having been apparently exhausted, we will pass on to the next paper, "Changes in Torpedo-Boat Designs," by Mr. Charles P. Wetherbee.

Mr. Wetherbee read his paper.

CHANGES IN TORPEDO-BOAT DESIGNS.

By Charles P. Wetherbee, Esq., Member.

[Read at the ninth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 14 and 15, 1901.]

Before recommending changes that are bound to reduce the speeds now attainable in smooth water, I wish to record for the Society data of what has been done and what can be done in the way of speed by torpedo-boats and destroyers of the present type.

Modern torpedo-boats have displacements ranging from 50 to 300 tons with indicated horse-powers varying from 700 to 6,000. Destroyers range from 300 to 475 tons with from 4,500 to 8,500 I. H. P. Both boats and destroyers have models with a ratio of length to beam varying from 10.5 to 9.67, and the ratio of depth to beam is between .65 and .75. The coefficient of fineness varies from .38 to .48.

Table 1 given below shows the results of carefully made progressive trials on the torpedo-boat Biddle, designed and built by the Bath Iron Works, Bath, Maine. The observations only included 29 knots speed, so the figures beyond this speed must not be taken as absolute. These results are also shown graphically on Plate 48. The trial data are accurate to 2 per cent. The tank results were supplied by Naval Constructor Taylor, U. S. N.

The Biddle's leading dimensions are:—

Length on trial L. W. L., feet and inches	157– 0
over all, feet and inches	157– 4
Beam at deck, feet and inches	15– 6
at water-line, feet and inches	16– 2¾
extreme, feet	17– 0
Depth amidships from keel to crown of beam, feet and inches	10– 9
Molded depth at side, feet	10– 0
Average draught of hull on trial, forward, feet and inches	4– 8½
aft, feet and inches	4–11½
mean, feet and inches	4–10
Mean displacement on trial, tons	168
Tons per inch immersion	4.4

Moment to alter trim 1 inch, foot tons	38.4
Area of trial water plane, square feet	1892
Center of gravity of trial water plane abaft midship section, feet.	7.25
Center of buoyancy abaft midship section, feet	1.50
above base, feet	3.09
Transverse metacenter above center of buoyancy, feet	4.96
base, feet	8.05
Center of gravity above base (trial condition), feet	6.10
Transverse metacentric height (trial condition), feet	1.95
Block coefficient for displacement	.478
Midship section coefficient	.724
Coefficient of trial water plane	.743
Wetted surface, square feet	2540
Diameter of propellers, feet	6.68
Pitch of propellers, feet	10.88
Projected blade area, each, square inches	1440

The condition of weights when the progressive trial took place was as follows:—

Hull complete, tons	47
Machinery complete with water	91
Reserve feed water	3
Trial load	27
Displacement	168

This vessel's hull is as light as it can safely be made. The machinery is about 16 tons heavier than is absolutely necessary to supply the 3,900 I. H. P. maintained on the two-hour sea trial. This excess of 16 tons is partly due to the fact that the evaporators and distiller with their pumps and the ash ejectors and steam heating system, though not belonging to propelling machinery proper, are included in the machinery weights, partly because the main engines are larger than is absolutely necessary, and partly to other causes to be mentioned later.

For our purposes, the weights can, therefore, be re-distributed as follows:—

Hull complete	47
Machinery and water to working levels, tons	75
Trial load, tons	46
Displacement	168

TABLE NO. 1.

PROGRESSIVE TRIAL OF BIDDLE.

Speed.	Revolutions per minute.	Per cent. slip.	Advance per revolution.	Revolutions to go 1 knot.	I. H. P.	Wave and eddy H. P.	Skin friction H. P.	Effective H. P.	Effective thrust in pounds.	Propulsive coefficient.	Admiralty coefficient.	Midship section coefficient.
A	B	C	D	E	F	G	H	I	J	K	L	M
11	117.	12.61	9.526	638	220	30	65	95	2810	.432	183.43	343
12	127.	12.17	9.574	635	286	45	85	130	3520	.507	183.19	343
13	137.	11.79	9.615	632	355	75	105	180	4500	.507	187.64	351
14	147.5	11.76	9.618	632	435	100	130	230	5350	.529	191.25	358
15	158.	11.75	9.620	632	522	130	160	290	6300	.556	196.03	367
16	169.5	12.25	9.565	636	630	180	190	370	7530	.587	197.13	369
17	181.5	12.93	9.491	640	760	245	225	470	9000	.618	196.	367
18	194.5	13.97	9.378	648	928	325	265	590	10670	.635	190.55	357
19	207.7	14.97	9.269	656	1138	420	305	725	12430	.637	182.74	342
20	220.	15.49	9.212	660	1370	500	355	855	13900	.624	177.05	331
21	231.4	15.64	9.196	661	1600	585	405	990	15350	.619	175.49	328
22	242.	15.49	9.212	660	1835	665	465	1130	16730	.616	175.94	329
23	252.4	15.29	9.234	658	2080	750	530	1280	18120	.615	177.35	332
24	262.6	15.04	9.261	656	2346	840	590	1430	19400	.610	178.66	334
25	273.	14.87	9.279	655	2636	915	670	1585	20650	.601	179.72	336
26	283.3	14.68	9.300	654	2932	1005	740	1745	21850	.595	181.75	340
27	294.	14.63	9.306	653	3257	1080	830	1910	23000	.586	183.23	343
28	304.2	14.44	9.327	652	3572	1165	920	2085	24250	.584	186.33	349
29	314.8	14.37	9.335	651	3910	1255	1015	2270	25480	.581	189.12	354
30	325.2	14.24	9.348	650	4225	1340	1120	2460	26700	.582	193.76	363

While the performance of this vessel is particularly good, the derived curves are characteristic of all torpedo-boats. Some points deserve special notice. First of all, every vessel has a certain critical speed where the character of several of the curves changes. This speed is fairly well represented for vessels of this model by the formula, Speed $= 9\sqrt[6]{\text{Displacement}}$, given by Mr. A. Normand. For the Biddle it is about 21 knots. By reference to the curve of "Midship Section Levels," it will be noted that until 21 knots is reached the vessel sinks bodily with relation to the plane of sea level, while at the same time changing trim; but at 21 knots, it begins to rise again, although it never reaches its still water height. We will refer to the speed where this change occurs as the "rising speed."

The curve of Admiralty coefficients, $\frac{D^{\frac{2}{3}}S^3}{I.H.P.}$, has a maximum at 16 knots, and a minimum at the rising speed. Beyond the rising speed it steadily increases in value, and no speeds have yet been reached where it ceases this upward tendency. The part of the curve below 21 knots is of the usual form found for large ships. Their corresponding full speed usually comes upon the falling part of the curve between 16 and 21 knots. For this reason, increased displacement of ships at ordinary speeds gives a much better performance with the same per cent. of displacement devoted to power. Their corresponding speeds for the same actual speed decrease as their dimensions increase, so that the Admiralty coefficient increases.

With torpedo-boats having their full power speed *beyond* the rising speed, this advantage largely disappears, owing to the change in slope of this curve. The lower values of the Admiralty coefficient below 16 knots are due to running large engines at low power. The curve of propulsive coefficients also shows this.

The propulsive coefficient happens to have a maximum value at the rising speed in the Biddle's case, but I regard this as accidental rather than as representing a general law. This maximum propulsive coefficient also happens to be at the speed where the apparent slip of the propellers is greatest.

The slip curve, however, is entirely characteristic. There is a maximum at the rising speed in the case of all boats with which the writer is familiar, and the apparent slip steadily decreases beyond this speed until the point of cavitation is reached. The speed where this occurs on the Biddle is about 29.5 knots.

The indicated horse-power curve shows that for the last 5 or 6 knots the increment per knot is nearly constant, being about 320 horse-power.

This is a characteristic of torpedo-boats when driven well beyond the rising speed, and it is due to this that such high speeds can be attained.

From the results obtained with the Biddle, curves of horse-power and speed have been computed by Froude's method for vessels of similar model having displacements varying from 40 tons to 600 tons, by 10-ton intervals.

A propulsive coefficient of .59 was used throughout. (Plate 2.) Across these curves are "limit" curves, showing the limiting speed possible in the present state of engineering knowledge with different trial loads varying from 0 to 120 tons. These curves are computed, using the weights of hull and machinery given in Tables 2 and 3, which follow:—

TABLE NO. 2.

DISPLACEMENT AND MINIMUM WEIGHT OF HULL IN TONS.

Displacement.	Hull.	Displacement.	Hull.	Displacement.	Hull.	Displacement.	Hull.	Displacement.	Hull.	Displacement.	Hull.
40	10.9	140	37.9	240	76.3	340	120.0	440	167.8	540	219.0
50	13.6	150	41.4	250	80.5	350	124.6	450	172.8	550	224.6
60	16.3	160	45.0	260	84.7	360	129.3	460	177.8	560	229.6
70	19.0	170	48.7	270	88.9	370	134.0	470	182.8	570	234.9
80	21.7	180	52.5	280	93.2	380	138.7	480	187.9	580	240.3
90	24.4	190	56.3	290	97.6	390	143.4	490	193.0	590	245.7
100	27.1	200	60.2	300	102.0	400	148.2	500	198.1	600	251.1
110	29.8	210	64.2	310	106.4	410	153.1	510	203.3		
120	32.5	220	68.2	320	110.9	420	158.0	520	208.5		
130	35.2	230	72.2	330	115.4	430	162.9	530	213.7		

TABLE 3.

MACHINERY WEIGHTS.

1 Boiler, Single Screw.						2 Boilers, Twin Screw.					
Sustained indicated horse-power.	Weight of boilers complete, with all accessories, stacks, uptakes, ash pans, grates, brickwork, valves, mountings, lagging, water in boilers and ash pans.*	Main engines complete, with shafting and propellers, in tons.	Valves, piping, manifolds with their lagging and clothing, strainers, condensers and water, circulating pumps, floor plates, ladders and gratings, oil and waste tanks, in tons.	Total machinery weight, in tons.	Weight per indicated horse-power, in pounds.	Sustained indicated horse-power.	Weight of boilers complete, with all accessories, stacks, uptakes, ash pans, grates, brickwork, valves, mountings, lagging, water in boilers and ash pans.*	Main engines complete, with shafting and propellers, in tons.	Valves, piping, manifolds with their lagging and clothing, strainers, condensers and water, circulating pumps, floor plates, ladders and gratings, oil and waste tanks, in tons.	Total machinery weight, in tons.	Weight per indicated horse-power, in pounds.
N	O	P	Q	R	S	N	O	P	Q	R	S
500	7.6	5.0	3.8	16.4	73.5	1500	18.3	11.6	7.6	37.5	56.0
750	9.2	5.8	4.4	19.4	57.9	1750	19.6	12.6	8.3	40.5	51.8
1000	10.7	6.8	5.1	22.6	50.6	2000	21.4	13.6	9.1	44.1	49.4
1250	12.2	8.0	5.7	25.9	46.4	2250	22.9	14.8	9.8	47.5	47.3
1500	13.7	9.5	6.4	29.6	44.2	2500	24.4	16.0	10.6	51.0	45.7
1750	15.3	11.6	7.0	33.9	43.4	2750	25.9	17.0	11.3	54.2	44.1
2000	16.8	13.9	7.7	38.4	43.0	3000	27.5	19.0	12.0	58.5	43.7
						3250	29.0	21.1	12.8	62.9	43.4
2 Boilers, Single Screw.						3500	30.5	23.2	13.5	67.2	43.0
1000	15.3	6.8	5.5	27.6	61.8	3750	32.0	25.5	14.3	71.8	42.9
1250	16.8	8.0	6.2	31.0	55.5	4000	33.5	27.8	15.0	76.3	42.7
1500	18.3	9.5	6.9	34.7	51.8	4250	35.1	30.1	15.7	80.9	42.6
1750	19.6	11.6	7.6	38.8	49.7	4500	36.6	32.4	16.5	85.5	42.6
2000	21.4	13.9	8.3	43.6	48.8						
2250	22.9	16.2	8.9	48.0	47.8						
2500	24.4	18.7	9.6	52.7	47.2						

* This weight also includes auxiliary feed pumps and blower engines and blowers, in tons.

TABLE 3—Continued.

MACHINERY WEIGHTS.

4 BOILERS, TWIN SCREW.						6 BOILERS, TRIPLE SCREWS.					
Sustained indicated horse-power.	Weight of boilers complete, with all accessories, stocks, uptakes, ash pans, grates, brickwork, valves, mountings, lagging, water in boilers and ash pans.	Main engines complete, with shafting and propellers, in tons.	Valves, piping, manifolds with their lagging and clothing, strainers, condensers and water, circulating pumps, floor plates, ladders and gratings, oil and waste tanks, in tons.	Total machinery weight, in tons.	Weight per indicated horse-power, in pounds.	Sustained indicated horse-power.	Weight of boilers complete, with all accessories, stacks, uptakes, ash pans, grates, brickwork, valves, mountings, lagging, water in boilers and ash pans.	Main engines complete, with shafting and propellers, in tons.	Valves, piping, manifolds with their lagging and clothing, strainers, condensers and water, circulating pumps, floor plates, ladders and gratings, oil and waste tanks, in tons.	Total machinery weight, in tons.	Weight per indicated horse-power, in pounds.
N	O	P	Q	R	S	N	O	P	Q	R	S
4000	42.7	27.8	17.9	88.4	49.5	8000	76.3	61.0	35.0	172.3	48.2
4250	44.3	30.1	18.7	93.1	49.1	8250	77.8	63.6	36.0	177.4	48.2
4500	45.8	32.4	19.6	97.8	48.7	8500	79.3	66.2	37.0	182.5	48.1
4750	47.3	34.9	20.5	102.7	48.4	8750	80.8	68.8	37.9	187.5	48.0
5000	48.8	37.4	21.4	107.6	48.2	9000	82.3	71.4	38.9	192.6	47.9
5250	50.3	39.9	22.3	112.5	48.0	9250	83.9	74.1	39.9	197.9	47.9
5500	51.9	42.4	23.2	117.5	47.9	9500	85.4	76.8	40.8	203.0	47.9
5750	53.4	45.0	24.1	122.5	47.7	9750	86.9	79.5	41.8	208.2	47.8
6000	54.9	47.6	25.0	127.5	47.6	10000	88.4	82.2	42.8	213.4	47.8
6250	56.4	50.3	25.8	132.5	47.5	10250	89.9	84.9	43.7	218.5	47.8
6500	57.9	53.0	26.8	137.7	47.5	10500	91.5	87.6	44.7	223.8	47.7
6750	59.5	55.7	27.6	142.8	47.4	10750	93.0	90.5	45.7	229.2	47.7
7000	61.0	58.4	28.6	148.0	47.4	11000	94.5	93.4	46.6	234.5	47.7
7250	62.5	61.3	29.4	153.2	47.3	11250	96.0	96.3	47.6	239.9	47.8
7500	64.0	64.2	30.3	158.5	47.3	11500	97.5	99.2	48.6	245.3	47.8
7750	65.5	67.1	31.2	163.8	47.3	11750	99.0	102.1	49.5	250.6	47.8
8000	67.0	70.0	32.1	169.1	47.3	12000	100.6	105.0	50.5	256.1	47.8
8250	68.6	72.9	33.0	174.5	47.4	12250	102.1	107.9	51.5	261.5	47.8
8500	70.1	75.8	33.9	179.8	47.4	12500	103.6	110 9	52.4	266.9	47.8

* This weight also includes auxiliary feed pumps and blower engines and blowers, in tons.

The weights given in the tables are minima and should not be used for design, unless one is thoroughly experienced in light construction.

The following machinery arrangements were used in computing the limit curves:—

Up to 1,000 I. H. P., one boiler, single screw.
1,000 to 2,500 I. H. P., two boilers, " "
2,500 to 4,500 I. H. P., " " twin screw.
4,500 to 8,500 I. H. P., four boilers, " "
8,500 to 12,500 I. H. P., six " triple screw.

Each engine has its own condenser. No arrangements contemplating a number of boilers other than an integral multiple of the number of engines are considered, because in them the difficulty of distributing the feed water becomes so great, when the evaporation is at all rapid, that they are impractical.

The steepness of the limit curves shows how little is gained in the weight of the load carried at any speed by an enormous increase in displacement. In 400-ton vessels at 30 knots, the displacement has to be increased 50 tons to carry 10 tons more load, while at higher speeds with larger vessels the displacement has to be increased as much as 100 tons for the same increase of load. As anything over the minimum in either hull or machinery weights is really load in the sense used here, the reason for keeping the displacement small and saving on the weights of hull and machinery, particularly the latter, becomes apparent.

In order to use the curves for any design it is simply necessary to add whatever the hull and machinery will weigh in excess of the figures given in the tables to the trial load, and then use the curves with this fictitious trial load to determine the displacement and horse-power necessary to fulfill a given programme.

For instance, let us assume that we have to determine the displacement and horse-power necessary to carry a trial load of 52 tons at a speed of 29 knots. The extra trial load, consisting of reserve feed water, evaporator, and distiller plant, ash ejectors, steam heat, weight due to any unusual requirements in the machinery, such as extra thick boiler tubes, extra thick and extra number of pipes, etc., is 33 tons, in addition to the usual trial weight of 52 tons specified for ordnance and ordnance outfit, coal, crew, boats, anchors, equipment, etc. Our trial load, therefore, is 85 tons. With the eye we follow up on the sheet of power curves an imaginary line half-way between the limit curves of 80 and 90 tons load, until this imaginary line crosses the 29-knot ordinate. The displacement of the nearest power curve

to this spot is 360 tons, which is the required displacement. From Table No. 2 the weight of hull corresponding to this displacement is 129.3 tons. We now decide that this hull is 30 tons lighter than we care to build, so adding 30 tons to the previous trial load gives us 115 tons for the new load, and returning to the curves with this trial load, we find the following dimensions fulfill our original programme as laid down :—

Displacement	480	tons
I. H. P.....	8,450	
Trial load.....	52.0	
Weight of hull.....	30 + 187.9 = 217.9	
Weight of machinery and feed.....	33 + 178.8 = 211.8	
Discrepancy	1. 7 tons	
	481.7	

The last two items are obtained from the preceding weight tables, adding the excess weights mentioned above.

When one becomes familiar with this simple method it can be operated in much less time than it has taken for this description.

The preceding pages and data apply to the type that has been adopted by all nations for torpedo-boats and destroyers. It is essentially a smooth water vessel and should be used for the purpose for which it was originally designed, *i. e.*, harbor defence.

If these vessels are to be sent to sea, a radical change in model is necessary. A vessel with a ratio of length to beam of 10 to 1, having the comparatively small dimensions of a torpedo-boat or destroyer, cannot be considered of a seaworthy model, nor is it one that can be made sufficiently strong to go to sea in bad weather. Additional material worked into the long shallow hull does not strengthen in proportion to the weight added. The result of this addition is simply a great reduction in speed for only a slight increase in strength, while the seaworthiness of the model is as poor as before. The dead wood aft is usually cut away to give a smaller tactical diameter. On this account the vessel will not steer in a seaway at low speed. If the speed is increased sufficiently to steer, the shocks of the sea become too great for the strength of the hull. Even neglecting the question of increased strength, it is doubtful whether a faster boat would not be obtained by decreasing the ratio of length to beam. The model that shows the least resistance in the tank is not necessarily the one which will give

the greatest speed when built into a boat, because the shape may be such as to require an abnormally heavy hull to get even the small strength necessary for smooth water work. All saving in machinery weight due to using a model requiring small power for a given speed, is more than offset by this extra hull weight, the result being that a model less efficient from a propulsive point of view, but having a lighter hull, goes faster.

The writer is familiar with two types of boat built to fulfill absolutely the same programme. One has a ratio of length to beam of 10½; the other has a ratio of 9⅔. The tank shows that the smaller ratio gives a loss in speed of nearly 7 per cent. for the same indicated horse-power at the same displacement. The latter boat, however, when built, owing to a smaller ratio of length to beam, had a considerably lighter hull with the same strength as the former, and was enabled to carry a greater trial load; and, in addition, to carry enough extra horse-power not only to make up this handicap of model, but to actually give three knots more speed and still have about 20 tons less displacement than the boat with the 10½ to 1 ratio. It is not fair, however, to credit the model with the entire difference between these boats, but a large amount is certainly due to it.

It is probable that this ratio of length to beam (9⅔) could be reduced even more with an advantage in speed; certainly, there is no doubt but that it would make the vessel more seaworthy, and, except perhaps at the extreme speeds, more efficient from a propulsive point of view. At the reduced speeds at which the vessel is run nine-tenths of the time during her existence, the smaller ratio of length to beam is amply sufficient, and the reduced ratio of wetted surface to displacement is such that the indicated horse-power for speeds where the skin resistance is a considerable part of the total resistance, would be less with the same displacement.

This large ratio of wetted surface to displacement explains why the Admiralty speed coefficient for these long lean boats never exceeds 200, even in conjunction with a fairly good propulsive coefficient. The model is finer than necessary for the speed where the maximum Admiralty coefficient occurs.

For small vessels designed to go to sea a ratio of length to beam of 8 to 1 is certainly none too small. This would probably cause a reduction of 25 per cent. in the smooth water speed, as compared with that of a model having a 10½ to 1 ratio at the same displacement. This disadvantage would entirely disappear in a seaway, and, in addition to having a hull better proportioned for strength, the vessel would be of a fit model to keep the sea in bad weather. The ratio of depth to beam (.66) at present in vogue should

not be decreased. With the ratio of length to beam of 8 to 1, and the same depth to beam ratio which at present prevails, the present scantlings for vessels of the same beam will give a fairly strong hull, if means are adopted to make the longitudinal strength members, particularly the deck, capable of developing their full strength in compression without buckling.

Unlike big ships, these small, high-powered vessels develop their greatest bending moment in sagging when the ends are supported on the crests of two adjacent waves with the middle in a hollow. The bending moment under these conditions for a vessel having a 10 to 1 ratio, is nearly three halves of that found when the conditions are reversed, and when the shock which results from jumping the vessel at high speed into a fairly heavy head sea is added to this bending moment, the compressive stresses in the upper longitudinal members are by far the largest that have to be taken into consideration. As the strength of the deck against buckling in compression depends upon the moment of inertia of the cross section and the modulus of elasticity of the steel, and not upon its compressive strength, the futility of the frequent attempts to increase the strength of this class of vessel by simply using stronger steel without increased scantlings, becomes apparent when it is remembered that the modulus of elasticity varies only 3 per cent. from twenty-nine millions throughout the range from the softest flange steel to the hardest tool steel.

Another frequently made mistake caused by this same constant value of the modulus of elasticity of all steel is the allowance in calculations of a fictitious thickness for stronger steels when used in conjunction with mild steel in tension, after which the augmented area so obtained is credited with the full strength of the weaker steel. This is wrong. The two classes of steel have the same modulus of elasticity and only offer resistance in proportion to their actual areas when subjected to stresses within their elasticity limits. What actually occurs is that the weaker steel in this combination passes its elastic limit first, thus throwing nearly the entire stress upon the remaining stronger steel with disastrous results.

In order to obtain sufficient compressive strength in the deck, other means than the mere thickening of the plates must be resorted to if a respectable degree of lightness is to be retained. To this end, longitudinals of angle, bulb angle, or, wherever possible, of plate, should be worked under the deck not over two feet apart in a manner similar to that by which the bottom of the boat is reinforced, and a 1½ inch non-fireproofed wood deck thickly fastened to the steel is also of great advantage in preventing the deck plate from buckling, also solving the problem of giving a deck which offers a proper foothold in a seaway.

The nearer the midship section of a vessel approaches a tube in shape, the stronger the vessel will be, and all longitudinal members are stronger in compression if given as much curve as possible in the athwartship direction. As great a crown of deck as is compatible with its use is desirable for this reason. Flat surfaces should be avoided everywhere. They are weak under all stresses except direct tension.

In addition to the unsuitable model, the torpedo-boats and destroyers of the United States Navy possess some faults peculiar to themselves. They have too many useless fittings and attachments. Their machinery, through complication, is too liable to break down. Both hull and machinery have unnecessary weight devoted to the unimportant details that would serve better if expended on the vital parts.

It is unfortunate that officers connected solely with torpedo-boats and destroyers in each of our designing bureaus in Washington are not usually on duty there long enough to follow a design made by them through the various stages of construction to the completed vessel. The torpedo-boat suffers more than large ships from this system. The lightness of touch and line of thought necessary to design and construct a successful torpedo-boat are instantly lost if the same man has charge of large ship work at the same time. It would also be of advantage to the design of a torpedo-boat as a whole if some one man of great experience with them could have the final decision concerning all matters entering into the design, no matter which Bureau has cognizance of the same. Many articles now creep in that are unnecessary and some things even are duplicated through being specified by two Bureaus.

This man should also be able to take a bird's eye view of the vessel as a whole, irrespective of department, and at the same time should be able to see in his mind's eye everything as it will finally be installed in each individual department.

Space is very limited in these boats, and although each article in itself, as its drawing comes up for approval, does not look to be much, the combination of all the individual articles specified, drawn, and approved, one at a time, frequently presents great difficulties owing to lack of space, and, if installed, they are inaccessible.

It is stated that a certain great shipyard in Europe has a high salaried official whose sole duty it is to go over the ships built by that concern and make a list of what can be left off in future vessels. There is certainly a great field for him on board our torpedo-boats and destroyers.

The following should be omitted from the hull and equipment as being out of place on vessels of this class :—

Complicated drainage systems.

W. T. magazines with their flooding, draining, lighting and ventilating apparatus for *fixed* ammunition.

The entire system of bells, push buttons, and enunciators, with their accompanying wiring in conduit and distribution boxes.

All voice pipes except from forward conning tower to engine rooms and to after conning tower.

The system of wiring in conduit and the type of fixtures for the electric plant.

In the steam engineering department of torpedo-boats and destroyers there are too many units of all descriptions, too many compartments; in some cases, too many boilers and too many engines, too many cylinders to an engine, too many blowers, too much pipe, too many valves, and too many ways of doing each particular manœuvre with the piping. The weights are pared too fine on the main engines, boilers, and blowers, while the piping, valves, fittings, manifolds, and other auxiliaries are no lighter than on any other class of ship. Pipes need not have the same allowance in thickness for wear as is given to a battleship as the vessels are shorter lived.

The engineer staff allotted to torpedo-boats is far too small, as will be seen by comparing the number of men allowed with those assigned to any other naval vessel of the same horse-power. This is unfortunate, for no other type of machinery deteriorates so rapidly from neglect or requires more expert supervision.

With this small staff in view, everything should be made as simple as possible. As we have to admit once for all that these vessels are not fit for sea-going work, the evaporators and especially the distillers with their accompanying piping and pumps, should be dispensed with and be replaced by reserve feed-tanks. Ash ejectors with their piping are useless.

Four-cylinder main engines take more care and room than those with three cylinders, besides using more steam. The same power is much more easily cared for and is lighter if generated in a small number of large boilers than in a large number of small ones.

Almost no attention is paid to the economy of coal consumption. The prevailing idea that the lightest machinery can be obtained by making as light an engine as possible without regard to steam economy is erroneous, for the penalty is paid when it comes to the boilers. They are far heavier than would be necessary if a little more weight were devoted to the engines to obtain a more economical use of steam.

The above faults are in all cases not errors in design, but they simply

result from the fact that these things are inappropriate on a harbor defence torpedo-boat, while they are absolutely necessary on a sea-going battleship.

The solution of the so-called torpedo-boat difficulties seems simple enough. The features criticised in our present boats are largely the outcome of the attempt to use the boats in a service for which the type was never originally intended. They must be restricted to harbor defence, and a new type more in the nature of a despatch boat must be designed for sea-going work.

These new boats must be safe at sea in any weather, and the larger ones (from 400 to 600 tons) must be able to make good speed in anything short of the worst weather. Unlike the torpedo-boat, which must be invisible above all things, these boats must have enough freeboard to make them seaworthy. They should have sufficient deadwood aft to make them hold their course at low speeds in bad weather, and if a small tactical diameter is deemed necessary, add a bow rudder to get it, especially in the single-screw boats. They must have a ratio of length to beam of at most 8 to 1, with a ratio of depth to beam of .7. Make the hulls amply strong and if the vessels are of 200 tons or under, use the composite construction instead of steel plating for shell and deck. After a strong, seaworthy hull and ample *necessary* equipment and ordnance are provided, I would turn my attention to getting as much *simple reliable* power into them as possible with the necessary coal and water. These vessels must have evaporators, but I do not regard distillers or ash ejectors as necessary.

As to machinery, I should use a *reliable* type of bent tube water-tube boiler and rugged three-cylinder triple-expansion engines throughout the class. The engine cylinder ratio should not exceed 5 to 1, as these boats will run most of the time at reduced power.

I would never use two engines where one would do. Providing for emergencies in the way of break downs, by duplicating moving parts in the smaller classes of boats, is more likely to be the cause of these same break downs. As boats having less than 1,000 horse-power will hardly be constructed for this service we can neglect any powers up to that amount. The arrangements for powers above 1,000 would be as tabulated on page 8, which I repeat here for convenience.

1,000	I. H. P.	to	2,500	I. H. P.,	2	boilers,	single	screw.
2,500	"	"	4,500	"	2	"	twin	"
4,500	"	"	8,500	"	4	"	"	"
8,500	"	"	12,500	"	6	"	triple	"

These powers are the forced draught power, and they could be main-

tained for only about four hours. The cruising power would be about half of the maximum.

Triple screws are adopted for the larger powers, as their engines fit into boats of these dimensions better than large twin screw engines, and at the same time the proper propeller area can be obtained without excessive draught of water.

Simplicity should be the motto throughout the design, and nothing that can possibly be dispensed with should be incorporated into either hull or machinery.

In conclusion, the writer wishes to acknowledge his indebtedness to many friends throughout the country who have supplied data. Particular thanks are due to the Bath Iron Works for access to their very complete torpedo-boat data and for the great assistance afforded in the preparation of this paper.

He is also under obligation to the Bureau of Construction and Repair, and to Naval Constructor D. W. Taylor, U. S. N., for the data obtained by towing the Biddle's model in the experimental tank.

APPENDIX.

Since writing this paper, the writer has been asked on several occasions concerning the effect of the condition of the bottom of a torpedo-boat upon its speed, and as the Bath Iron Works has some accurate data on that subject, it is appended here as probably being of interest.

A progressive trial was made of the torpedo-boat Barney on December 1, 1900, when she had been overboard 126 days, lying in the water of the Kennebec river, which is brackish and having a maximum density at high tide of about 1.013. During this interval of 126 days, the Barney was run out at sea about fourteen times, the trips averaging not over six hours each. She was hauled out immediately after the trial and the bottom was found to be very rough, but there were no barnacles on it, but the paint had risen into pimples about the size of the end of a lead pencil and they were very close together, there being no smooth places.

The Barney's displacement on the trial was 168 tons. This data can therefore be properly compared with that obtained on the Biddle, which is given in the first part of the paper. The Biddle is an identical boat, but the paint was in good condition and the condenser scoops projected on a transverse plane, showed about 20 square inches less area. The Biddle had been overboard 35 days when the progressive trial was made.

The following is a table of speed, revolutions and indicated horse power for the Barney, and those for the Biddle have been repeated here for convenience in comparison. The results are startling, to say the least.

Speed.	Biddle.	Barney.	Biddle.	Barney.
	Revs. per min., clean bottom.	Revs. per min., dirty bottom.	I. H. P., clean bottom.	I. H. P., dirty bottom.
12	127.0	127.0	286	313
13	137.0	137.4	355	396
14	147.5	148.7	435	489
15	158.0	160.5	522	602
16	169.5	172.6	630	745
17	181.5	185.2	760	927
18	194.5	198.0	928	1150
19	207.7	210.9	1138	1410
20	220.0	223.4	1370	1705
21	231.4	235.2	1600	2002
22	242.0	246.0	1835	2290
23	252.4	256.6	2080	2585
24	262.6	266.7	2346	2892
25	273.0	277.0	2636	3230
26	283.3	287.5	2932	3590
27	294.0	298.0	3257	3960
28	304.2	308.4	3572	4340
29	314.8	318.2	3910	4730

DISCUSSION.

THE CHAIRMAN (MR. MCFARLAND):—Gentlemen, I am sure you have all been interested very much in this paper, and I must say that if the object of a paper is to stir up a lively discussion, this paper is likely to fulfil that purpose. I think Mr. Wetherbee certainly had the courage of his convictions—I am not saying whether I agree with everything he said or not—but it certainly seems to me that there are enough people here interested in torpedo-boat design to make a very interesting and very lively discussion. Mr. Magoun is here. We would like to have his criticism.

MR. HENRY A. MAGOUN, *Member*:—The point that strikes me particularly is the part of the paper relating to what can be left out. I think that anybody having anything to do with these boats has run into that at one time or another, and the uppermost thing in my mind is a little matter that we ran into the other day relating to the interior communication system. I give this to illustrate what can happen. We took up with the proper parties the arrangement and requirements for the wiring conduits and junction boxes and we quite innocently went ahead and ordered more or less of the stuff and among it the junction boxes. Very fortunately we got one of the junction boxes before the order was filled, and, on weighing it, and looking into the matter we found that we had about nine hundred pounds of junction boxes on a little torpedo-boat destroyer. We managed to cut that down to three hundred pounds, and I really believe that is three hundred pounds too much. I merely give this to show what you can get into very innocently, indeed. In the same connection, I may say that on the boats to which I refer there is about fifteen hundred feet of speaking tube, which, I think, weighs a pound to the foot, and we found that some of it could be very easily done away with from the fact that when a man is standing in the conning tower, there is a distance of only about ten feet between him and the place the tube was led to. Of course, there are a great many other instances that could be quoted, showing almost the same thing in other parts of the boat.

THE CHAIRMAN:—I take it, then, that you are in quite general agreement with Mr. Wetherbee?

MR. MAGOUN:—I think I am.

ADMIRAL BOWLES:—I would like to be in the beginning of the discussion, because I do not think it expedient to be drawn into the general discussion of torpedo-boats at the present time. However, it is not out of place for me to express my satisfaction at the reading of this paper, and also my sincere compliments to Mr. Wetherbee upon the matter that he has laid before the Society. It is timely and necessary and proper. It is a case of very intelligent hind-sight, based upon the most careful application,—and intelligent application,—of his ideas to his own work, and very successful work, too. It shows the necessity, undoubtedly, of the continuous application of some experienced man to this subject. The criticisms that he has made are so uniformly severe upon all parts of the Navy Department that there is no hard feeling, I am sure. (Laughter and applause.) But the paper is not only one of criticism. He has included in this paper the actual results of his experience towards building up the subject anew, and I wish to express dissent from the basic idea of the new design which he offers in order that we may not be led astray in what we may do in the future. It seems to me that a vessel for the Navy which can only be used for harbor defense is absolutely useless. The torpedo-

boats have got to accompany the fleet, and we have no use whatever for vessels that are limited to smooth water service. These torpedo-boats have got to go to sea—even those that we have.

LIEUTENANT CHANDLER:—I trust the Admiral will say that they have got to come back.

ADMIRAL BOWLES:—I sympathize entirely with the wish of the author for a man in Washington who has a final decision irrespective of bureau, and I hope that he is growing up. That feature has impressed itself upon me in the examination of the torpedo-boats and torpedo-boat destroyers that are building, and the effort is being made to supply that necessity with regard to the boats that are now being completed.

THE CHAIRMAN:—I notice my distinguished friend Mr. Hyde down at the rear of the room. He has built some torpedo-boats lately, and has had some valuable experience with them. Perhaps he can give us the benefit of his experience.

MR. JOHN S. HYDE, *Member*:—Whatever experience I have had in the matter is recorded in Mr. Wetherbee's paper. I don't think I can add anything to it. His paper was based on the performance of the boats that we have built, and I assure you that he knows much more about the subject than I do. I don't know that he made one correction in his paper and that was this: As I remember it, he gave two days in which the Barney and Bagley were tried, and through an error in the paper it was given incorrectly. One boat was tried on the forenoon of one day; the other had the official two-hour run in the afternoon of the same day, and, on the morning of the next day, the first boat had a two-hours' official trial. So that practically we had about three trials in thirty hours.

THE CHAIRMAN:—Mr. Forbes, do you care to contribute to this discussion? The question has been raised about the complication of the engine and that sort of thing. As you are an expert on small high-speed engines, perhaps you may have something to say.

MR. FORBES:—Mr. Chairman and gentlemen, the paper is extremely interesting to me. I don't know why it is, but the whole Anglo-Saxon race seems to delight in killing something, and if they cannot do that, in preparing something to kill with. The torpedo-boat is supposed to be a weapon of war, and a weapon of war must have the prerequisite of absolute certainty of action. When we lose that, we lose everything. The nastiest feeling that takes possession of a person is in not knowing whether his production is going to succeed or fail. Much worse must it be for anybody who is in charge of a mechanical contrivance where he is pretty sure from the very start that something is going to go wrong. Torpedo-boats seem

to have the habit of going wrong at all sorts of times and in all sorts of ways. This must arise from one of three causes, it seems to me, or perhaps a fourth cause may be introduced. The design itself may be wrong, the workmanship and execution of the design may be wrong, or the materials may be unsatisfactory; finally, if design and construction are satisfactory and it is not handled properly it will fail. This paper evidently shows to me that we have gone too far towards lightness in our designs, that we cannot rely on what we make to carry out our work. Now it seems to me that if we are going to build torpedo-boats in the future that the fixing of responsibility must be settled. It is certainly admirable to have one man who will make a final decision. That is very fine; you get your decision, but it is not always right. I think that the Navy is doing its work most admirably, supervising everything in every way, and there is certainly in each department some individual who has the final say, as I understand it. There is no doubt that it is the auxiliary that destroys efficiency in nine cases out of ten; in the chase of the Itata it was the blower engine that was always giving out. It seems to me it is not the general design so much as the execution of the details which insures success; some little three-eighths inch nut drops off and the whole machinery breaks down, and it really seems that success lies more in the people in my position carrying out thoroughly well, without haggling and without dispute, the designs that are given them by the Navy Department. It is really in doing this that we are more likely to arrive at more satisfactory work in torpedo-boats than in a general change of design. The torpedo-boat referred to is the design, I believe, of Mr. Normand, who is undoubtedly the boldest builder of engines that we have. His work, I believe, bears a record of never having broken down. What Mr. Hyde has said bears that out. I do not think we have reached such a record in our own designs yet. The engines are certainly not pretty; they look like almost anything, but they are of first-class design, and they are beautiful in their results, and everything becomes beautiful when it does its work. I wish to say, finally, that if the designs of the Department are honestly carried out, and the work thoroughly well done, and they live right up to the specifications, we will have far less trouble with our torpedo boats after they are completed.

LIEUTENANT ALBERT P. NIBLACK, *Associate*:—I think that anyone who has followed the torpedo-boat discussion and papers through the transactions of this Society will agree with me that, if we plot the curve of their value technically, Mr. Wetherbee's paper will be found to correspond with the highest point up to the present time. There is, fortunately, a disposition to follow out in practical application in the design all the best ideas. These papers, of course, are more or less unsettling, because the important point, from a strictly technical standpoint, of the officer who has to handle these torpedo-boats, is that we do not really need the best if we use to the best advantage what we have. Of the thirty boats which we now have, there are at least seventeen distinct designs. That is in itself a misfortune, and the

trouble is very aptly emphasized in the fact that there is not one single individual who is responsible. Consequently, any criticism is legitimate, because there has been a lack of co-ordination of ideas, not only in what torpedo-boats are to be used for, but in the different elements in their design. For instance, we are told now that most of the torpedo-boats we have are smooth-water boats. One of the considerations which we have had to confront in designing has been, by utilizing canals, to transfer boats from one base to another base through the interior line of communication which we are so very fortunate as to have. This, of course, when accepted as a limitation, has changed somewhat the design of the smaller boat. Mr. Wetherbee made a statement about the triple screw engine, and it has just occurred to me that the Italian government had three boats designed in 1887, with triple screws, the Tripoli, the Montebello, and the Mont St. Bernard, and they have been in operation all this time, but we hear very little of them. The triple screw, I believe, originated in the United States. The general question of the application of all these ideas to design is, as I said before, on the purely theoretical consideration that it would be wise to follow out all of them. What we want is a large number of boats that we can use in general torpedo-boat service, and we will not get very far in practical results until we try thoroughly what we have.

THE CHAIRMAN:—We have with us another gentleman who is fortunate enough, or unfortunate enough, to have noted some of these little changes. We will be glad to hear from Lieutenant Chandler.

LIEUTENANT L. H. CHANDLER, U. S. N.:—In reading this paper and in hearing it read I gained a good deal of information, and, in the main, I am in agreement with what Mr. Wetherbee says. I think an especially valuable thing he has shown here is on page 172, in the paragraph about the middle of the page, where he shows the vast increase of displacement which must be given to a vessel with high speed, in order to carry a very small useful weight, pointing out from this fact the necessity for keeping down superfluous weights. Now, it seems to me, without venturing to criticise the past action of anybody, that we have sought for standardization. I believe that Mr. Niblack is very fond of it. We have it in some ways, and in other ways we have let it go. We have not always been happy. For instance, I think we have stuck to the standardization of a great many fittings, and I think it has been unfortunate and has resulted in adding unnecessary weight; the tendency has been to lighten torpedo-boats, not by lightening their fittings, but by taking weight out of the hull and engines. I think it might be better if we started on the other tack and put in lighter fittings. I think there is one principle that ought always to be kept in mind in trying to save weights in this way, and that is, as I have heard Mr. Wetherbee express it, although he did not introduce such a statement in this paper, the best way to save weight is not so much by cutting down the size of any particular individual thing, as it is in making that

individual thing strong enough to make it add strength in other and useful directions. I also think that in our past and present construction we have not kept enough in our minds the distinction between smooth water and rough water. Most of the boats have got to go to sea, as Admiral Bowles says. I do not think they have got to go with the fleet; but the question of harbor defense, as expressed, is, perhaps, rather misleading. A harbor-defense boat is a boat that has to operate around a harbor, and should not only be able to keep foreign ships out, but also should be able to go outside and drive them off. As soon as a torpedo-boat gets outside of a harbor it doesn't really make much difference whether she is five miles from the coast or fifteen miles or fifty. If there is bad weather, she must be able to go outside, if necessary, as I understand it. So I think the harbor-defense boats should be equal to that. I think that designs have been approved for boats which run successful smooth-water trials, without giving enough consideration to what they would do when they were out in a little sea. There are two or three little details which occur to me as exemplifying the truth of this. For instance, the open hot-well proposition works beautifully, and is very satisfactory on trial runs; but when you get out in a seaway, it throws all the feed water out in the bilge, which is not satisfactory. In the same way, I am a little doubtful about the scoop condenser. It scoops all right in smooth water, but if you get out in a moderate sea, for instance, and you are headed to it and not going very fast, the water does not come in the scoop very well. I found that the mouth of the scoop was alternately lifted above the water slightly and deeply buried, and I think that unless some other means of overcoming this defect is provided, such an arrangement is very unsatisfactory. I think for that reason our boats ought to be designed more with the rough-water service in view, even if they have a comparatively short radius of action.

As to the question of decreasing weights by leaving out things, I think we all realize that there is a great deal of this that might be done. In one paragraph, Mr. Wetherbee alludes to the engineer's force allotted to torpedo-boats. Simply make the comparison between the force I have on board the Bailey at the present time and the force that Mr. Wetherbee had on the trial trip. He had 56 men in the engine-room force. He also spent considerable time, I suppose, before the trial, training them. I have on the Bailey 30. When they came aboard last summer there was not one of them who had ever before seen a water-tube boiler or a torpedo-boat engine. I am not certain that my presence below would have brought the work to perfection, as I did not come up through the water-tube business myself, unfortunately for me; but the dissimilarity of the trial trip and service conditions is obvious. I notice that Mr. Wetherbee says that ash ejectors are probably useless. I have found that a good way to get rid of ashes at sea is to get them all ready, have a little whip handy and dump them on the deck alongside of the hatch; that was all we had to do with them. We had more trouble in keeping the men there than we did in making the ashes go. To go back again to the rough and smooth water

question and the question of steering; if you want a boat which you can handle quickly, that will turn readily in her own length, you have got to go to our present type, which is the cut-away flat-bottom aft; a boat of that description goes around very quickly and is very handy. But if you take the same boat out to sea, especially with a following sea, she goes around very nicely, too, but without any assistance from the helmsman. Recently we left New York to go to Port Royal; I told the other officer who accompanied me that we took our departure from Sandy Hook Lightship and stood south; practically it was within six points of it, and the compass was doing four points more on its own responsibility, so I was not sure of our locality until we fetched up at Port Royal. Hatches that are essentially tight in smooth water are not necessarily so in rough water. A hatch to keep the rain out won't keep the sea out. Bunker and other plates that are tight in a smooth-water run, cease to be water tight if the ship works a little; and dozens of fittings, such as rail stanchions and awning stanchions that are strong enough in themselves for the work they have to do, give way when a big man is thrown against them in a seaway. It seems to me we must consider these things and make them strong enough for sea service; but I do not see how the extra weight can help resulting in a diminution of speed. I think we have gone beyond the effective sea speed; that when we get our thirty knots we are coming very near the smooth-water basis; if that is what we want we have it in certain cases. But I think we had better drop high speeds and put in enough extra weight in scantlings and fittings to make the boat thoroughly sea-going.

As to the proportion of length to beam, I am not enough of a naval architect to figure that out. I will let Mr. Wetherbee settle that with some of the "X" people.

NAVAL CONSTRUCTOR D. W. TAYLOR, *Member of Council*:—I should like to say a word or two about this valuable paper. I notice that Mr. Wetherbee has raised one point concerning which we are old antagonists, and, as I think there is something to be said on the other side, I will make a few remarks. What I speak of will be found on page 175. Mr. Wetherbee objects to the use of stronger steel in conjunction with the ordinary mild steel. In the particular case to which he refers, the combination was that of nickel steel and mild steel in adjacent longitudinal members in the engine and boiler space. Of course, the nickel steel has greater ultimate strength and a much higher elastic limit than the other material with which it was associated, and Mr. Wetherbee's point at first sight seems to be well taken. But when you consider the conditions existing, I think there is something to be said in favor of the combination noted. In the first place, the deck of the boat between the bulkheads is a good deal cut up with companions and hatches; the strength is very much reduced naturally, so that you cannot count on the full area of the section; you must count on big holes here and there which take away a large part of the strength. In addition to this, the midship part of the boat, in the vicinity of engines

and boilers, is more subject to the vicissitudes of heat. If you take a piece of steel and raise its temperature about 55 degrees Fahrenheit, it will stretch as much as if you gave it a tension of 10,000 pounds to the square inch. The natural result is that, in case of tension (which has to be considered as well as compression), if you have steam up, the central part of the boat is already somewhat stretched, so that there is a reasonable supposition that the nickel steel and the mild steel will here reach their elastic limit in tension very possibly under similar conditions of stress. Of course, the stresses to which Mr. Wetherbee refers are, and I think he will agree with me, largely conventional; we cannot determine by placing a boat on the crest or hollow of a wave what the exact strain is. It is very largely a matter of convention, and we must be governed in this respect very much by practical experience. So far as I know, in practically every instance in which torpedo-boat destroyers or larger vessels have given trouble as regards strength, it has been in the deck, and not compression or tension. We have had very little experience in that line ourselves, because we have just begun building such vessels, and accurate information with regard to foreign boats is difficult to obtain. But I know of several instances which are fairly well authenticated, in which the upper decks of these boats have given a great deal of trouble, as in the recent case of the Cobra, which broke in two. There is some question as to what gave way first, but there is very little doubt as to the fact that, when the Admiralty thought it necessary to strengthen her they put seven tons in the deck and not down below, which goes to show that they considered that the deck required the greatest strengthening. Mr. Wetherbee has prepared some diagrams to show some of the results of the model basin experiments, and I think these results for boats of this type, at this extreme speed, have a good deal of interest and show how fallacious some of our old ideas with regard to resistance were. I have taken the liberty of preparing a diagram, Plate 50, which reproduces the curve of propulsive coefficient, what Mr. Wetherbee would term the slip per cent. There were two curves given in Mr. Wetherbee's diagram; one was the curve of effective thrust, and the other was the curve of indicated horse-power, which Mr. Wetherbee took from the progressive trials. We used to think a few years ago that the resistance varied about as the square of the speed and the horse-power as the cube of the speed. So what I have done is to calculate the powers according to which the resistances were varying and to plot the results as curves. You will see that the index of power starts in about $3\frac{3}{4}$ at 11 knots, falls down to about $3^{3}/_{10}$ at 14 knots, climbs up again to over 4 at 17 knots, and then falls off steadily the rest of the way up to 30 knots. The I. H. P. index starts in a little below 3, is a minimum at about 14 knots, rises to a maximum at about 18 knots, and then proceeds to fall off until, at 29 knots, that index is about $2^{4}/_{10}$, showing that the resistance instead of varying with the square of the speed was varying as the one and four-tenths power of the speed. That shows one reason why these boats drive so easily at the higher speed. A peculiar feature about this is that either curve taken alone would throw some doubt on the result.

We would not suppose initially that these indices would describe these curved lines, but you will see that the same general features are shown both in the E. H. P. curve and the I. H. P. curve which are entirely independent; that the I. H. P. shows a tendency to have the same humps, but not so pronounced. We also used to think that the propulsive coefficient was constant; that the same fractional power was always devoted to propelling the ship. We know now that this is not the case. As a matter of fact, even if this curve were somewhat inexact, we know that its general character is correct. Of course the propulsive coefficients used in making the curve were calculated for the bare hull, and the actual propulsive coefficient would be somewhat affected by the scoops, etc. In trying to find out the reason for the peculiar features of the curve, I came to the conclusion that they were due almost entirely to the longitudinal distribution of the waves which a boat makes. We find that at 12 knots there is a bow wave which, for convenience, may be taken as directly at the bow, and the next crest of the wave comes just amidships, and the next crest, of course, comes at the stern. The result is that at 12 knots the bow wave is reinforcing the stern wave and you have the maximum wave disturbance and the resistance increases more rapidly. That is the actual fact from observation of the distribution of the waves along the side of the model and that is reproduced exactly to all intents and purposes on both sides of the ship. As the speed increases these waves lengthen and this crest (pointing to the sketch) moves aft, and it so happens that the length of this wave at 17 knots speed corresponds to the length of the ship. In other words, this bow wave can reinforce the stern wave at 17 knots. Of course, at a certain speed between 12 and 17 knots, probably at about 14 knots, the hollow at the bow wave combines with the crest of the stern wave and we get the minimum interference, and we will naturally expect the curve of resistance to fall off. In the actual curves, we really get the humps and hollows; these are very marked in the larger ships. But in the curve of the torpedo-boats, we do not as a rule get the humps and hollows; we get little flats on the curve. You will see that the curve has a maximum at 17 knots and a minimum at about 14 knots. It proceeds to fall off after that, and the reason is there are no more crests to travel aft. What happens is that as the speed is increased beyond 17 knots this hollow moves aft more and more and tends more and more to pull down the stern wave, and you do not get another lump on that curve. You do not get another coincidence of the crests of the bow and stern waves. There is another little point in this connection which shows the accuracy of these results. The curve of slip per cent., Plate 50, is apparent slip, not actual slip, but it bears a pretty fair relation to actual slip; it will be observed that it reaches a maximum at about the point where these curves of index reach the value 3. In other words, we all know that the thrusts of the propeller as the speed increases vary approximately as the squares of the speed, and if that were absolutely true at the index 3, we should find (supposing the coefficient also remains constant and varies very little) that the slip would remain constant; it will be noted that it comes to a horizontal line and tends to remain constant; also, as the index falls below value 3 the slip falls off.

THE CHAIRMAN:—Has any other gentleman any comments to make?

MR. LINNARD:—Mr. Chairman, I have only a slight comment to make. I have been engaged recently in seeing how this simplification of torpedo-boats could be accomplished. Like a great many others I am very fond of simplicity, but I find that simplicity generally consists in leaving out the other fellow's things and adding a few of your own, so I am afraid it is not so easy, as it might appear on the surface, to leave out superfluous things, because everyone is interested in making his own part secure and letting the other fellow shift for himself. We all know that any structure of this kind is the result of many minds working together, and that no one can fully grasp all the details of the whole, and the tendency is to multiply these details which you think on your part are necessary and every one engaged in the operation does likewise, and the result is an accumulation of fittings which, when you glance at them as a whole, appear very unnecessary. It can only be remedied by having some reconciling authority of broad views to exercise arbitrary control over the omission or the diminution of these units. The question of the relative dimensions utilized in Mr. Wetherbee's curves is one that has occurred to me. It would appear that, as I understand Mr. Wetherbee's curves, they are deduced from similar models. Now, as the absolute dimensions derived from the model increase, all the dimensions of the ship increase, presumably, proportionately, and we know that in that case the linear dimensions and the metacentric heights are also increased and, as you proceed to the smaller ships, are diminished proportionately. At the same time, the tendency would be, in the smaller vessels, to increase the height of the centre of gravity, still further decreasing the metacentric height, and I would ask Mr. Wetherbee whether any account was taken of that fact or whether it was within the limits considered as immaterial. Mr. Wetherbee advocates shortening these ships and proposes thereby to get an increased speed. It reminds me a little of the Irishman who was offered a stove that would save half his fuel and he said he would buy two and save the whole of it. I would like to know whether Mr. Wetherbee considers that this limit of eight to one is the least that is possible, or whether he would go further under certain conditions.

MR. THEODORE LUCAS, *Member*:—I should like to draw attention to the curve submitted by Mr. Taylor, showing the index for effective horse-power, that a maximum occurs at 12 knots when the ship is precisely the length of the trochoidal wave, and that the minimum of the index occurs at about 14 knots, where the length of the boat is one and one-half times the length of the trochoidal wave; in other words, the maximum occurs when the length of the ship is precisely the same as the length of the trochoidal wave, and that there is a maximum when the ship is twice the length of the wave, while the minimum occurs when the ship is one and one-half times, and is quite likely to occur at two and a half and three and a half times the length of the wave. Apparently, there are no humps or hollows

any more. I think a general law can be deduced from that, and I formulated it in a published article. It was derived from Froude's experiments. We all remember those famous experiments having the length of a ship composed of eighty feet fore-body and eighty feet after-body. Froude took this body and afterwards inserted middle body, and noted the humps and hollows in the wave resistance, which always occurred at an even multiple of the length of the ship, while the minimum occurred at one and a half, two and a half, three and a half times the length of the ship. I think an approximately correct general law can be deduced from that which will give a good indication of where the constant hollows would occur. I would lay more stress upon the index of effective pressure in this connection, because it is far more accurate, I believe, in the tank, than it is possible for the indicated horse-power to be, since a good many varying quantities will affect the actual trials of the ship. In the tank experiments we will see the thing pure and simple. The resistance is only skin and wave resistance, while, in the actual trials a good many other considerations will enter—efficiency of the propeller, and a good many others that may obscure the result. In Mr. Wetherbee's paper, he speaks of the form being as nearly circular as possible; that that gives the best strength. Now, I believe in our torpedo-boat design we are limited quite a good deal in draught, and it is often a question if it is not perhaps better to take a little fuller midship section and a flat deck and gain a little more inside room as well as greater displacement near the midship portion of the boat and make up for the lack of strength, which Mr. Wetherbee states always exists in flat surfaces, by a little additional material and stringers, which would stiffen the vessel. I believe in many cases these round sections have given too little displacement to the ship, considered as a whole. Under the draught limitations, it is a question whether additional displacement would not often give the desirable additional stability which, in some of the boats of the French navy, we have sometimes seen sadly lacking.

THE CHAIRMAN:—Colonel Stevens, have you anything to say?

COLONEL STEVENS:—The only question that arose in my mind was one that was suggested by some experience with these flat stern vessels—whether any of the officers who handled them had any experience in having them towed in a seaway. My own experience has been that, with such vessels, even with very small sea, as soon as the propeller stops turning it is almost impossible to steer them at any speed at all. I should think that, in a seaway, if one of those vessels became unmanageable, she would have to look out for herself; no consort could take her in.

THE CHAIRMAN:—I would like to ask Mr. Wetherbee to give us still further information, in the reply which he will make in a few moments, on that portion of his paper where he speaks about the importance of having the main machinery as simple as possible and getting rid of extra appendages, etc. Of course, I think, in general, that we all agree with that statement, but, in another place, the author referred to

the importance of trying to get economy in the engine, not to simply drive the life out of the boilers; we all know that this brings up the question of difference between Mr. Normand's methods and those of the rest of the torpedo-boat builders. Normand has always laid great stress on the economy of the engine. The others have always used a very simple engine, and not a very economical one, and have forced the boiler enough to get the desired speed. I would ask Mr. Wetherbee to state whether, in getting this extra economy in the engine, some of this simplicity would not have to be sacrificed to some extent. I ask this because, from my recollection of what I have read about Mr. Normand's method of attaining economy, there were certain additions to the engine that nobody else had ever used; so it would seem as though the two statements were somewhat inconsistent. I am sure that we all feel that the paper itself is most interesting, and the discussion that has taken place has been most instructive.

MR. WETHERBEE:—Mr. Chairman and gentlemen, of course my gauge of the value of a paper is the extent to which it provokes discussion, whether such discussion is *pro* or *con*. It is the discussion we are seeking, and I think it will be best for me to answer a few of the questions that have been asked without expressing any opinion on the greater part of the discussion that has taken place. Of course, I wish to thank Admiral Bowles for his very kind reception of the paper, but in stating that he dissents from the basic idea of the new design, I think that he is laboring under a misunderstanding. I do not propose that any more boats, solely for harbor work, be designed. I think it is safer for the lives of some of our men if we keep those we now have for harbor defense and build a type of sea-going torpedo-boat; this is the boat I recommend. In my opinion, the ratio of eight to one in length to beam is all that can safely be used in a small boat. To put my ideas in a few words, I propose that we continue to design torpedo-boats in this country, but that they be built for sea-going work; that we obtain the sea-going qualities by reducing the ratio of length to beam, with consequent increase of strength, and by increasing the scantlings—both of these changes are at the expense of speed in smooth water, but the result will give us sea-going boats which can accompany the fleet.

I am gratified to note the way in which the members who have spoken agree with me on this matter of having some one individual responsible for decisions on all matters relating to design and fittings. We have found this to be quite true in our own experience. Concerning Mr. Chandler's remarks regarding scoops for the condensers, the condenser pipes, if properly fitted, did not seem to give any trouble in any sea, if the pipes were so fitted as to go through the shells very obliquely instead of being normal to the shell and depending on the scoop on the outside to force the water through. In fact, I have grave doubts if the scoop is necessary at all under those conditions. When a boat is laboring in a seaway and the scoops come out occasionally, or where she is being run at a power very much below her

maximum power,—under these circumstances the probability of getting a proper vacuum is greatly reduced below that when running at full power.

Concerning Mr. Taylor's remarks about the nickel-steel in the midships section, which he illustrated, if he does not agree with me that the greatest stresses are compression stresses in the deck, all my argnment falls to the ground. I cannot agree with him on that question. But it strikes me that, even with the midships section Mr. Taylor indicated, if you neglect the center of the deck entirely the vessel is too weak even with nickel-steel.

Referring to Mr. Linnard's question about the metacentric height, there has been very little difficulty in obtaining sufficient metacentric height since we departed from what might be called the tooth-pick stern model. Since we have adopted the broad flat stern, the metacentric height, even on a narrow boat with such a stern, is enormous. For the Normand model, which we adopted, the metacentric height is 1.9 feet loaded, and this should be ample for all requirements. Of course, using a smaller ratio of length to beam, which I advocate, would go to increase the metacentric height in all sizes of boats, provided the weights are the same; it is a very easy matter to reduce the metacentric height, we all know.

Concerning the remarks of Mr. Lucas about the square midship section and the flat deck, I can only say that in most torpedo-boats the propellers dip anywhere from two feet to four and a half feet below the keel, so I cannot see any objection to giving them considerable dead rise. Concerning the stability of the French torpedo-boats, the boats that gave them trouble were of an older model, before this round section was adopted and before the flat stern was used to such an exaggerated extent. The boats have a very wide range of stability. Stability curves show that the stability vanishes at an angle between 90 and 100 degrees. Of course, the fastenings for the boiler and the machinery would probably give way before the boat heeled over that far. So I think they are all right.

In reply to Colonel Stevens regarding towing, we have had considerable experience in towing these boats, and the Navy Department has also done some towing under very difficult circumstances. You all probably know that the bow of the Craven was badly smashed by collision with the Dahlgren a year ago, the first two compartments being bent over and set at right angles to the center line of the vessel. The big wooden fender around the Dahlgren saved her from being cut in two. The Craven was taken back to the torpedo station at Newport, as the accident happened within six or eight miles of that place. It was decided to tow her to Portsmouth for repairs. The boats were originally supplied with a shackle in the bow which was fastened into cleat plates riveted to the sheer strakes. At first it was decided to tow the boat stern first. I advised against that, as the boats are very low at the stern, and suggested instead that the boat be towed by securing the tow line to this shackle in the bow, believing that a pull on the tow line would straighten the bow out. These boats are very much like an accordeon and the scheme worked beautifully. The boats were towed from Newport to Portsmouth without any difficulty

whatever, in the fall of the year, when the weather was not very fine. In our own experience, we have been in the habit of sending the boats from Bath, Maine, to Portsmouth, New Hampshire, a distance of 72 miles, the vessels being towed the whole way. The Bagley and Barney were sent together in this way. They have towing shackles, as before described, in the bow and a ring in the stern. A hawser was secured to them; the propellers were blocked so that the engine would not turn and cut the cylinder. Under these conditions we found that no steering whatever was necessary, the propellers being so far aft and the pull of the tow line being just at the bow served to keep the boat in a perfectly straight course, and made her follow the tug beautifully. I must say, however, that the engineer of the tug (who had recently towed two of those large six-masted schooners) said that those two boats pulled harder with the propellers blocked than the six-masted schooners did.

In reply to Mr. McFarland concerning the economy of fuel and the arrangement of the engines which permits us to obtain this economy, I do not think that any complication is involved. I was under the impression that the method used was generally understood, or I would have explained it more thoroughly before. There is a separator on the main steam-pipe, at the throttle of each engine. From the bottom of this separator a pipe leads to the top jacket of the high-pressure cylinder. I might add, in passing, that all the cylinders are jacketed, and that they are all jacketed with boiler steam on both heads and bottoms, with the exception of the low-pressure cylinder, which is jacketed with steam from the intermediate steam chest, which has a maximum of about 100 pounds on the barrel; if boiler-pressure steam were used for the low-pressure cylinder, it would necessitate excessive thickness of cast-iron barrel. This separator before noted delivers this mixture of water and steam at the bottom, through a pipe that runs to the jacket on the high-pressure head. This jacket drains to the high-pressure barrel-jacket, which in turn drains to the high-pressure bottom jacket. This in turn drains into the jacket of the head of the intermediate cylinder, which drains down through the barrel jacket and then to the head of the low-pressure, and from the head of the low pressure to the bottom of the low, not going into the barrel jacket for the reason I previously gave. The steam then goes into the separator and is used to drive the blower engine. That is the secret of the operation, because, otherwise, you could not get your steam and water to run up hill, as it had to do in several places, through the engine, except by having a heavy current of steam going through. This also has the advantage of providing a constant current of fresh steam through the jackets, and avoiding any air pockets, which are liable to occur in steam jackets fitted in the ordinary way; and this steam finally drives the blower engine, and we can turn the exhaust of the blower engine again into the low-pressure chest, if we care to, or into the condenser; so it is seen that the steam has to do its work thoroughly before it gets through.

THE CHAIRMAN:—I had an impression that Mr. Normand went into the matter of compression very fully; that he had some sort of safety valve on the steam chest to avoid getting too great pressure in it.

MR. WETHERBEE:—He uses a very high compression, but there is no complication connected with it, because the valve has been in use by many other people, and it simply consists in having a relief valve in the steam port, a relief valve from the cylinder into the preceding steam chest, so that if you get compression above the chest pressure this valve, which is only loaded with a very light spring, will lift and let the cylinder compress back into its own steam chest and that serves the function of a relief valve. There are no relief valves on the cylinders, so that there is no complication involved there. There are ordinary relief valves on the steam chest of the intermediate and low, which you would have in any case. Then he uses a feed water heater which I do not think is very complicated. The steam for heating it is taken from the low-pressure steam chest; the feed water is pumped from the feed pumps, which are coupled to the engine, through the heater, to the boiler. This feeder is fitted with tubes through which the feed water is pumped and the tubes are fitted with retarders, and, with the ordinary steam pressures usual in the receivers of those boats (from eighteen to twenty-one pounds), the feed will be heated to 200° or 240° as it goes into the boiler, so that this heater is remarkably efficient. I might say that the heater has one foot of surface for every 15 horse-power, so that they are not large. In conclusion, I wish to thank all the gentlemen who have entered into the discussion for the pleasant way in which they have received the paper.

NAVAL CONSTRUCTOR LINNARD:—I move that the thanks of the Society be extended to Mr. Wetherbee for this interesting and valuable paper.

The motion was unanimously carried.

ADMIRAL BOWLES:—Mr. Chairman, I venture to ask you to permit me to interrupt the regular proceedings and bring before the Society a matter which seems to me of great importance. We are now beginning the tenth year of our existence and have reached a very gratifying membership, now about 820. Up to the present time the Society has been managed and its affairs conducted by a few devoted and unselfish souls at very small expense to the Society, and the result has been that we have been able to accumulate, a little at a time, a small amount of money—$8,000. Last year, for the first time in these ten years, our expenses exceeded our receipts and they bid fair to continue to do so. Now it seems to me we have reached a point where we have got to make a step in advance in order that the Society may advance in usefulness commensurate with its importance; not that these same few devoted souls have become selfish or anything of that sort, but they have grown older and their regular duties have increased and they feel that they are unable to continue to give the time that this Society requires; and it is an increasing amount of time, and I can assure you that the people who have managed the affairs of the Society have found it necessary, every day, to give a considerable amount of time to its affairs. You may not understand it, but it is absolutely neces-

sary. There are other matters in the management of the Society which are unsatisfactory. Its management has been confined to a very few people who take an interest in its affairs; you might say a narrow oligarchy. There has not been the proper interest taken in the election of officers, and these positions have not been made the legitimate ambitions for professional distinction which they should be. For these and other reasons, which I will not take your time to state, I have prepared this motion:

Resolved, That the Council be requested to revise the constitution with a view of increasing the revenues of the Society by an increase of dues and entrance fees, and also to recommend such change in the method of election of officers as may seem desirable, and to consider other matters which may appear to be for the advancement of the Society. Such revision, when made, to be submitted for the consideration and vote of each member and associate; the result of such vote to be submitted at the next annual meeting for action, with a recommendation of the Council.

Now, I wish to say in explanation, that the dues ought to be increased. They are now only $5 a year, very much less than those of any other society of corresponding standing, either in this country or abroad. I am not in favor, however, of increasing the dues of the junior members of this Society, upon whom the expense would fall heavily, and who might be discouraged from continuing with the Society on that account; but I am in favor of having the dues of members increased, and I am in favor of having the dues of a considerable number, if not all of the associates, increased. We all know, and need not conceal it from one another, that there are a great many people who join this Society for the business advantages afforded; there is no reason why they should not materially assist in the advancement of our profession, and I am sure they will generally be willing to do so.

With these remarks I submit the motion, Mr. Chairman, for your consideration.

THE CHAIRMAN:—Gentlemen, you have heard the motion made by Admiral Bowles, and the remarks that he has made. Has anybody any further remarks to make?

MR. STEVENSON TAYLOR:—I hardly think, in view of the remarks made by Admiral Bowles, that any further statement is necessary at the present time, beyond heartily seconding his motion. It seems to be very clear to every member of this Society that the dues must be increased. Certainly the members of the Council, at their meeting a few days ago, when they considered the financial report, came to that conclusion, and I do not see how it is possible for anyone to object to it. It seems to me the members and the associate members should have their dues increased, even if we allowed the dues of the junior members to remain as they are. One of the questions that arose out of the discussion at the Council meeting was the

distribution of our proceedings to the junior members. If I remember correctly, the cost of publication this last year was between seven and eight dollars per volume for the total membership, and, as the junior members only paid $3, they were getting a very large margin of profit. I have no doubt at all that the members and associates would rather allow the junior members' dues to remain as they are in order that they may have the full benefit of the Society for their future education. I am heartily in favor of the motion, and I trust it will be unanimously adopted.

ADMIRAL BOWLES:—I omitted one thing which I would like to make clear, and that is that my principal object in submitting this motion is that I consider it essential to the future usefulness of the Society that we should be able to employ a permanent professional Secretary.

THE CHAIRMAN:—I presume most of you know that the work of the Secretary heretofore, all the nine years of the existence of the Society, has been practically a labor of love. This cannot go on forever. You cannot always find people who will be willing to do this.

MR. F. M. WHEELER, *Member*:—I would like to make a few remarks on this motion. In the American Society of Mechanical Engineers, when I was on the Finance Committee, this same trouble came up. We were trying to run the Association too cheaply, and although they wished to keep the dues down as much as possible to encourage the growth of the Society, we finally got into financial difficulties, and it became necessary to do something. We were then paying $10 a year, and I made the motion to increase to $15. It was thought by a great many that that was too much. But afterwards it proved to be a most excellent thing. I am only surprised that we have so long gotten along as well as we have in this Society on such low dues. But I think we can all put our hands in our pockets and pay more, including the juniors. It costs money to publish these proceedings. In fact, we are practically in the publishing business. It is the principal item of expense, and we ought not to suffer a loss by doing that. The American Society of Mechanical Engineers when they increased their dues added 50 per cent. right away to their income. They were able to bind the volumes nicely without exacting an extra charge as before, and to do many things in a first-class way which added very much to the reputation of the Society. As regards the suggestion that was made by Admiral Bowles about having a paid secretary, I think that is very important. What is done in other societies ought to be adopted by us, and if we are to be as successful as some of our sister societies, the sooner we do it the better.

THE CHAIRMAN:—Gentlemen, you have all heard the motion and I presume you all understand it thoroughly. Those who are in favor will say aye, those opposed, no.

The motion was carried unanimously.

THE CHAIRMAN:—Now the Secretary asks me, gentlemen, to remind you that we have one great treat still in store for you, that is the banquet to-night; but some of you seem diffident and are not making your arrangements quite as promptly as is desired, so that you are all requested to make known your intentions as promptly as possible. There will be a meeting of the Council immediately before the afternoon session, and unless there is some objection we will now adjourn until two-thirty o'clock.

The meeting adjourned until 2.30 P. M.

SESSION OF FRIDAY AFTERNOON.

The meeting was called to order on Friday afternoon, November 15, at 3 o'clock, Colonel Edwin A. Stevens, Vice-President, in the chair.

THE CHAIRMAN:—The Secretary will read the report of the Council of the Society in relation to members, associates, and juniors recommended for election.

The Secretary read the following list:—

Members (8).

Charles C. Bowers, Superintending Engineer, Crescent Shipyard, Elizabeth, N. J.

Daniel Douglass Boyd Brown, Leading Hull Draughtsman, Wm. R. Trigg Co., Richmond, Va.

Goold H. Bull, Professor of Marine Engineering, Webb's Academy, Fordham Heights, N. Y.

Samuel Shearer Caskey, Superintendent of Machinery Construction, Harlan & Hollingsworth Co., Wilmington, Del.

Henry Robert Curwen, Chief Draughtsman, Crescent Shipyard, Elizabeth, N. J.

Robert H. Fraser, Engineer Surveyor, American Record, Dorchester, Mass.

Howard C. Higgins, Superintending Engineer, Old Dominion S. S. Co., New York.

Ernest A. Perham, Leading Draughtsman, Hull Dept., Fore River S. & E. Co., Quincy, Mass.

Promotion from Associate to Member (3).

Arthur P. Allen, Designer and Supervising Draughtsman, Moran Brothers Co., Seattle, Wash.

George C. Cook, Naval Architect and Engineer, 15–25 Whitehall St., New York.

Wm. Carr Nickum, Draughtsman, Wm. Cramp & Sons S. & E. B. Co., Philadelphia, Penna.

Promotion from Junior to Member (2).

Edwin S. Alexander, Inspector, Newport News S. B. & D. D. Co., Newport News, Va.

Ernest I. Cornbrooks, Draughtsman, Maryland Steel Co., Sparrow's Point, Md.

Associates (21).

John F. Bruton, Quarterman in Charge of Power Plant, C. & R. Dept., Navy Yard, New York.

Lieut. Lloyd Horwitz Chandler, U. S. N., Navy Department, Washington, D. C.

John P. Chase, Draughtsman, Fore River S. & E. Co., Quincy, Mass.

Daniel Cox, Draughtsman, Hull Dept., Neafie & Levy S. & E. B. Co., Philadelphia, Penna.

Frederick Pomeroy Hall, Hull Draughtsman, Designing Dept., Holland Torpedo-Boat Co., Elizabeth, N. J.

Frederick Hepburn, Leading Draughtsman, N. Y. Shipbuilding Co., Camden, N. J.

Charles E. Houston, Hull Draughtsman, Neafie & Levy S. & E. B. Co., Philadelphia, Penna.

Howard H. Hummel, Student in Naval Architecture, Webb's Academy, Fordham Heights, N. Y.

Wm. Elbert Jackson, Student in Naval Architecture, Webb's Academy, Fordham Heights, N. Y.

Jacob Katzenstein, Supt. L. Katzenstein & Co., 358 West St., N. Y.

James G. Lindsay, Senior Member of James G. Lindsay & Co., Philadelphia, Penna.

Walter Henry Millard, Designing Dept., Holland Torpedo-Boat Co., Elizabeth, N. J.

Alfred Wynn Minuse, Student in Naval Architecture, Webb's Academy, Fordham Heights, N. Y.

Axel E. Petre, General Manager, Midvale Steel Co., Philadelphia, Penna.

Lieut. Wilfrid Van Ness Powelson, U. S. N., Navy Yard, N. Y.

Walter H. Rogers, Leading Draughtsman, Hull Construction, Wm. Cramp & Sons S. & E. B. Co., Philadelphia, Penna.

Edson B. Schock, Superintending Construction, Holland Torpedo-Boat Co., Elizabeth, N. J.

Watson C. Schreiner, Student in Naval Architecture, Webb's Academy, Fordham Heights, N. Y.

Townsend J. Smith, Hull Draughtsman, T. S. Marvel & Co., Newburgh, New York.

Thomas A. Watson, President, Fore River S. & E. Co., Quincy, Mass.

Wm. I. Wyman, Draughtsman, C. & R. Dept., Navy Yard, Boston, Mass.

Juniors (10).

Charles C. Brush, Draughtsman, Neafie & Levy S. & E. B. Co., Philadelphia, Penna.

W. Benson Caswell, Draughtsman, Neafie & Levy S. & E. B. Co., Philadelphia, Penna.

Herbert Henshaw Christophersen, Student in Naval Architecture, Webb's Academy, Fordham Heights, New York.

Matthews Erastus Davis, Asst. Foreman of Ships, N. Y. Shipbuilding Co., Camden, N. J.

Harley Hichborn Grogan, Asst. Draughtsman, Office of Suptg. Constructor, Fore River S. & E. Co., Quincy, Mass.

Robert Stanley Haight, Asst. to Superintending Engineer, Old Dominion S. S. Co., New York.

Wm. Franklin Hussey, Hull Draughtsman, Neafie & Levy S. & E. B. Co., Philadelphia, Penna.

Edwin E. Mager, Hull Draughtsman, Neafie & Levy S. & E. B. Co., Philadelphia, Penna.

Samuel Deems McComb, Draughtsman, Hull Department, Townsend & Downey Shipbuilding & Repair Co., Shooters Island, N. Y.

Charles Albert North, Ship Draughtsman, Townsend & Downey Shipbuilding & Repair Co., Shooters Island, N. Y.

THE CHAIRMAN:—The question is now on the election of the gentlemen recommended by the Council.

MR. LEWIS NIXON:—I move their election.

The motion was put and unanimously carried.

THE CHAIRMAN:—The next proceeding in order is the reading of a paper on "Late Developments in Ordnance and Armor," by J. S. Meigs, Esq., Associate.

Mr. Meigs read his paper.

LATE DEVELOPMENTS IN ORDNANCE AND ARMOR.

By J. F. Meigs, Esq., Ordnance Engineer, Associate.

[Read at the ninth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 14 and 15, 1901.]

In the past ten years the developments in armor plate have consisted in, *first*, the introduction of nickel in the armor; *second*, the adoption of the super-carbonizing process invented by Mr. Harvey; and, *third*, the perfecting of this super-carbonizing process by Krupp. Parallel with these improvements in armor plate, whose tendency, or rather the tendency of the latter two, has been to produce armor of very hard outside face, the velocities of projectiles have advanced. From a purely rational point of view this would appear to be unreasonable, because hard armor is best attacked by heavy projectiles, which, if the size and weight of the gun remains constant, would mean a lowering of velocities. In other words, hard armor which may be cracked is best attacked by heavy projectiles, and comparatively soft armor by light, fast-flying shot.

In the ten years under review the advance in the resisting power of armor has been very material. The advance made by the introduction of nickel and by the adoption of the Harvey surface-hardening process, which were brought in at about the same time, added probably quite 25 per cent. to the power of armor plate of fixed thickness to resist penetration. The adoption of the Krupp process probably added another 10 per cent. to the resisting power of armor plate, and also added very materially to its power to resist cracking. It is perhaps not too much to say that the armor of to-day has 40 or 50 per cent. more power to resist penetration than that of 10 or 12 years ago of the same thickness. Its power to resist cracking is very difficult to measure numerically. This power, however, is very greatly increased, as much because of the better methods of melting and forging large masses of steel, as for any other reasons. It seems now almost impossible to crack well-made armor plates under conditions approaching those usual in reception tests. Their surfaces are spalled for an inch or two depth, and the tough back of the plate is very resisting to cracking effect. In a word, when comparing the armored ship of to-day with the ship armored

twelve years ago, say, with the same thickness, we find the former has an enormous superiority.

The same degree of superiority is not found in the gun itself. The guns of a fixed caliber are longer and heavier than they were, and the powder is better than it was ten years ago, but, as stated, the gun itself, though shaped differently, has not improved from improvement in quality of material so much as armor plate. Ten years ago a 6-inch gun weighed about 11,000 pounds, while now it weighs from 17,000 to 18,000. The former gun gave barely 2,000 feet with the brown powder then in use, while the later gun, with its charge of smokeless powder, will give 3,000 feet or more, and thereby its energy is more than doubled. This, however, it will be noted, is largely because the gun is heavier than the old 6-inch guns. The advantage which is most marked in the past decade in ship's guns, or, to use the more general term, in guns of position—that is, guns that are not moved from the place where they are set—is in rapidity of fire. Not many weeks since an English ship of war fired eight shots from a 6-inch gun in one minute, at a range of about 1,500 yards, at a target 15 feet high, and struck it every time. Such a feat a few years since would have been thought impossible, and, in fact, many people with whom I have conversed will not believe what I have stated to be a fact, but the reports that I read of it in the newspapers, which, of course, are available to anybody, are of such a nature that I myself believe its truth. No doubt the ammunition was close by, or it is possible that the ammunition supply arrangements may have been extremely good, and that the ammunition may have been brought from the magazine. This is unlikely, however, and in the statement that I have seen, this is not said to have been the case. The actual trial of how many complete rounds of ammunition can be delivered to each gun in a ship would appear to constitute as important a datum regarding her efficiency as her record on the measured mile. It would depend, of course, upon her structural arrangements and upon the skill of the ammunition passers.

No doubt, also, the ship's course was carefully selected in such a way as to blow the smoke away from the gun, for, although powder is now called smokeless, the heated gases coming from the gun render it impossible to aim properly until they have blown away, and, if blowing towards the firer, they would hurt his eyes so that he could not make good practice. We hear, also, of guns, whose projectiles weigh 350 pounds, being fired three or four times a minute. These things are no doubt true, in at least a limited degree, and constitute a very marked advance in gunnery, for, while it would not be possible to fire a 6-inch gun many minutes at eight rounds

per minute, both because the crew would become exhausted and because the gun and other parts would heat, yet the fact that this can be done for a short time, when it becomes necessary, is a matter of great consequence. It appears that this end has been reached, principally, if not solely, by putting it into the hands of one man—namely, the firer, to do everything that is required to aim and fire the gun, without moving and without taking his eyes from the line of sight. He simply keeps the gun aimed all the time, and fires it whenever it is loaded and as quickly as it is loaded. The adoption of measures of safety whereby he, standing half way from the breech to the trunnion of the gun, and with his back to the breech block, cannot fire the gun unless the breech is properly closed and the man handling it out of the way, are details of manufacture which have been mastered.

While, of course, it is very difficult to say at what range modern ships will fight, yet it would appear that this matter must be settled before any conclusions can be reached with regard to the kind of guns to be placed in ships. For example, if the old Pensacola, having ten smooth-bore 9-inch guns on her broadside, can choose her range, she is more than a match for the modern Atlanta, of equal tonnage, carrying three 6-inch and two 8-inch breech-loading guns on her broadside. The ship that can control range can outclass the other. It is necessary to determine to what range ships will usually come before determining upon their batteries and armor.

There is much difference of opinion in regard to the range at which the modern gun is effective. Possibly this comes from a neglect to appreciate the circumstances of the use of the gun, rather than the accuracy of the gun itself. The gun is itself very accurate, but the circumstances of its use are much less so. The changes in velocity, due to changes in powder, cause some variation in the range of the gun, but the greatest difficulty consists in the absence of correct knowledge of the range, and, in guns in ships, in the continual motion of the ship—both the rolling motion and the swing, when the ship is turning through an angle. My impression is that those who have witnessed a good deal of firing from guns in coast fortifications think that a ship would be in great danger from the guns of a fort in direct fire at 8,000 or 9,000 yards, and I assume that this presupposes that the range is accurately known, as it usually may be in a fort. I should think, however, that so great accuracy would be attained only when the firing is very deliberate, and when such elements as the muzzle velocity due to the powder in use are accurately known. In a ship gun, where the range is rarely accurately known, and where the ship is usually moving, accurate practice cannot be made at so great a distance as 8,000 yards. I think that most

naval officers believe that with their guns, and in reasonably smooth water, they can make fair practice at 3,000 or 4,000 yards. My own belief is that this is too great an estimate. At 3,000 yards range a target 15 feet high, when a gun is rolling at an angular rate of one degree per second, which is a fair rate only, the prolongation of the line of sight of the gun would sweep across the freeboard of a ship 15 feet high, which would include all the vulnerable part of the ship, in somewhat less than one-tenth of a second. This is a period of time too small to be appreciated by a man's nerves, and, indeed, the accuracy of operation of the firing apparatus of the gun can hardly come within this limit.

It is usual to assert that naval guns stand in greater need of high velocity than land guns, in order that the trajectory may be flatter and the gunner independent of the range at great distances. At present, for practical purposes, the trajectory is a straight line—having regard to the height of the broadside of the ship—for about 1,500 yards in length. By insisting, however, upon what is now known as "power" in guns,—that is, the use of elongated and fast-flying projectiles rather than those which approach a rounded form and move with lower velocity, much is necessarily lost in smashing effect. The shell charges become smaller and the mine effect of the shell becomes less, and the damage to constructions that are hit becomes less. The shot of the old-fashioned 9-inch smooth-bore guns weighed 90 pounds, while that of the modern 6-inch rifle weighs 100 pounds. For anything that it can penetrate the former constitutes a more dangerous projectile than the latter. Against the upper works of vessels the old round shot is by no means to be despised. This all arouses the question of what caliber of gun the designer of a warship will put in. He has a certain weight to put into battery, and the question comes to him at once:—Having 100 tons to dispose in guns and gun carriages, shall it be put in so many 8-inch guns or in so many 6-inch guns? Roughly, the weights of guns vary with the cube of the calibers, and for every 8-inch gun you can have eight 4-inch. As the 4-inch gun fires a good deal faster than the 8-inch, the weight of metal and of bursting charge thrown by the battery of light guns would considerably exceed that thrown by the heavier ones. The weight of a single discharge from a battery of fixed weight, of any caliber, would remain about the same, of course. These, it seems, are the great tactical questions in regard to guns:—First, having a fixed system of guns,—that is, having guns of differing calibers in which the dimensions are about proportional to the calibers, what size guns shall we use; and, second, there is the other question:—Is the system right? That is, should 8-inch guns

weigh what they do, or should they weigh more or less; or, to put it in another way, are we satisfied with present weights of guns, or do we want more velocity in a gun of fixed caliber, and are we willing to accept the increased weight that must be given to the gun to get this.

Much discussion is going on now in regard to the substitution of 7-inch and 7.5-inch guns for 6-inch guns on the broadside of battleships, and, while everybody must be desirous of increasing to the utmost the size of the guns, and, consequently, their destructive effect, it must be remembered that this inevitably makes the weight carried heavier, and in certain cases limits the speed of firing. It has been generally asserted that guns cease to be rapid-fire when increased beyond 6 inches in caliber. Even 6-inch guns must have their cartridges separate from the projectile, and, as the projectile weighs 100 pounds, it is about all that one man wants to put into a gun quickly when standing on the deck of a rolling ship.

Another matter of much interest just now, particularly in this country, is the disposition of the turrets in battleships. I had the opportunity last summer to go over the battleships Alabama, Kearsarge, and Massachusetts, when lying in Newport harbor. It is not necessary here to describe the batteries of these three ships, as they are all known to you, but there was perhaps no sentiment aroused within me more clearly defined than that of sympathy with the various officers and boards that have been called upon from time to time to pass upon the batteries of these ships. Volumes could be written upon the subject, and no one can reach a conclusion without first forming a clear idea of what sort of combat these ships are most likely to be engaged in. As we do not know what this will be, it seems that no one can dogmatize on the subject, and that all that can be done is that the matter shall be determined according to our lights, and that thereafter we should hope for the best. In passing, however, it may be noted that American ships always, from the first, have been very heavily gunned, and that again and again this has been shown, when the time of trial arrived, to be of vital importance.

There are many minor improvements which have been introduced in the use of guns, as, for instance, the almost general adoption of telescopic sights, which give greater accuracy of fire, the use of electric firing, which fires the gun more quickly when the sights are on, the use of automatic ejectors for the primers, and improvements in fuses. Also, high explosives have come to be used very generally in shell. These explosives are usually of the picric acid order, but have not yet attained much acceptance in this country. Many believe that they are quite as safe in shells as the old

black powder, and, as they certainly break shells into a larger number of parts than the older explosives, it would appear that they are certain to be adopted in time. Also caps have been used on the point of armor-piercing projectiles, and these are becoming universal. Many theories are advanced as to why they help a shot through an armor plate, but of the fact that they do so help it, there is no doubt whatever. It is said that they are least effective in oblique fire, and, while this may be so, it is yet a thing very hard to demonstrate, as the penetration of all armor-piercing projectiles falls off very quickly when the shell does not strike fairly point first. The cause of this seems to be largely that the whole back end of the projectile, upon striking a plate obliquely, is wrenched off, and the very second that a projectile, when striking an armor plate, begins to go to pieces, that second it is done for and its chance of doing any material harm to the armor becomes nil.

In no respect, perhaps, has greater improvement in gunnery been made in the last ten years than in the practically universal adoption of smokeless powder. At present, the pure nitro-cellulose powder is generally held to be much the best. Though smokelessness is usually put forward as the great advantage of the modern powder, yet the absence of solid residue in the products of combustion would appear to be the real gain in this regard. In the old powders about 40 per cent. of solid residue was projected from the gun as uselessly as though its weight was in sand, while in the modern powder all or nearly all the charge turns into gas and helps to propel the projectile. On the other hand, the gas of the modern powder, particularly where much nitro-glycerine is used in making it up, as is the case in cordite, for example, is of very high temperature and scores and gutters the gun very badly. This is much less so with the nitro-cellulose variety, and with the latter, where moderate pressures are used—say, of 34,000 or 35,000 pounds, the scoring of the gun is not a very serious matter. The scoring of steel by hot powder gas is a most extraordinary phenomenon. If there is ever so little a channel through which the gas can escape the steel is washed out and disappears in a most wonderful way. As, however, the propelling power of powder depends upon the amount of gas given off and upon the temperature to which it is heated, it is desirable, of course, to use as high a temperature as we can. In other words, if the scoring and guttering of the gas could be controlled, the hotter it is the greater the velocity attained would be. As a matter of fact, however, when high pressures—say, 40,000 or 50,000 pounds, are reached, the destructive power of the gas becomes very great. When the new powders came in first it was thought

that a charge of one-fourth or one-third the weight of the projectile would be ample to give all the velocity that would be needed, but, owing to the desire to attain very high velocities (3,000 feet and upwards), the weights of charges have been creeping up of late until now, in some guns, they approach one-half the weight of the projectile, which is the figure at which they stood roughly in the days of brown or cocoa powder.

It need hardly be pointed out here how important a matter this is as controlling the sizes of magazines of ships. Indeed, in some classes of artillery, as for instance in field artillery, where weight is of vital importance, there are many who believe that some form of nitro-glycerine powder may still be used, as it is with shoulder guns.

A word may now be said in regard to the improvements in the quality of steel in guns, and whatever is said about this may apply, it is obvious, to shafting. Ten or twelve years since or thereabouts, the tube of a large gun or a large piece of shafting which showed 40,000 pounds elastic strength was considered something rather out of the common; nowadays it is not at all unusual to be able to secure 60,000 pounds elastic strength in the same forging. This could be attained with about 16 per cent. elongation. Indeed, up to guns of about 5-inch caliber, the tube and jacket of which constitute pretty reasonably sized forgings, qualities as high as 70,000 pounds elastic strength, with 16 per cent. elongation, and about 130,000 ultimate strength, have been reached at Bethlehem. Nor is there any reason to anticipate that this material is any less safe to use than the softer material. Indeed, all the evidence goes the other way. It would not, perhaps, be desirable to reduce a gun's thickness of wall very materially in case this stronger material is used, but rather to consider the gained strength as so much surety against disastrous accident. Indeed it would be impossible to materially reduce the weights of guns without increasing that of their carriages, as with a lighter gun the recoil would become much more violent and harder to control. It would be very easy, if desired by ship constructors, to give the gun a larger base upon which to stand and to increase the length of recoil in many instances. As a striking example of this latter, it may be mentioned that modern quick-fire field artillery use a 3-inch gun 90 inches long, employing a recoil of 40 or 50 inches—that is, the gun recoils half of its own length. Of course it might be entirely undesirable to let a ship gun recoil through anything like such a distance as this, but there is nothing impossible about it if desired. Possibly one of the most striking changes made in gunnery in the last few years has been in the field artillery. By means of the long recoil stated, and by the adoption of the so-called spade at the end of the

trail, which spade enters the ground, these guns have become capable of a very rapid rate of fire and have been fired 10 and even 20 times within a minute. The men handling them sit on seats fast to the trail, and anyone who saw the field guns of a few years since, when in action, will realize what a revolution this means.

It used to be accepted as a sort of an axiom in ship design that the armor should be so distributed and the vulnerable parts of a ship so protected that no one shot could destroy her. This has led, in view of the heavy turret guns carried by ships, to the heavy side armor and heavy armor on the turret and barbette. I think a study of naval history will bear me out in the statement that the resistance of ships has been overcome nearly always by silencing the battery and not by destroying the ship. If this is so, a ship should carry large bored guns of respectable penetrating and mine power, and their magazines and ammunition supply arrangements should be such that they could be served at maximum rate. Roughly, if the systems of the guns are the same—that is, if the guns are of the same dimensions, when stated in calibers, then for every 12-inch gun we may have 27 4-inch guns, and as the rate of fire of each 4-inch gun, owing to the size of which the ammunition could be handled at the gun—providing it could be brought there from the magazine—is quite five times that of the 12-inch, it becomes apparent how much total metal and bursting charge is sacrificed for the heavy turret gun. Suppose a ship, instead of, say, 4 12-inch guns and 4 8-inch and 16 6-inch guns, should carry about 170 4-inch guns or 80 4-inch on her broadside, or a total of 56 6-inch, say. Which would be the better ship? Of course this parallel may be reasonably combated, but to me it means a great deal. If it be true that ships are overcome by their batteries being silenced—that is, by their men being driven from the guns—then the number of shots fired becomes a question of great importance.

At the present time it may be said that the principal guns of battleships are in turrets, usually in pairs, and are of 12-inch caliber. The thickness of the turret armor is 12 inches usually, or less. The so-called secondary batteries of battleships are composed of 6-inch guns, which are in broadside, and are often isolated by transverse armor and armor to their rears, which renders it unlikely that more than one of them could be silenced by the bursting of a single shell. These 6-inch guns are for the most part behind 5-inch and 6-inch armor. In other words, as the actual velocities for turret guns now afloat do not exceed an average of 2,300 feet, the turret guns of contending battleships can destroy each other only when within 600 or 700 yards range, and the guns of their secondary batteries, the ruling velocities

of these being put at 2,300 or 2,400 feet, when within 1,000 yards range. More powerful guns of both classes are now being built and mounted, but a great number of 6-inch and 12-inch guns now in use would come within the approximate rule just laid down. Thus we may fix the fighting range of battleships at 1,000 yards.

I think perhaps people might differ with me a little bit about the figures here, but I think the verdict has gone forth. I do not know where you will find it better expressed than in the battleship of to-day. The battleship of to-day cannot have her turrets penetrated except under the most favorable circumstances. We may talk about the range of fire of ships and other matters of opinion, but this is the best concrete fact that I know, namely, that battleships to-day cannot seriously injure each other (that is, guns of the light class, firing at the light armor) much outside of the very limited ranges above stated. Of course much depends on the class of armor. A great many battleships have what we call soft armor and a great many have comparatively low powered five-inch and six-inch guns, and the others have big ones; but, strictly speaking, I do not see how the guns of the secondary battery of battleships are going to penetrate the secondary armor of another ship unless she comes pretty close. They cannot even do it on the proving ground where the projectiles always hit the plate square; whereas, on board ship, the armor remains just as good as on the proving ground, and the gun is at a great disadvantage. It seems to me that the designers of battleships recently built and those now building must have concluded that ships are going to fight at about a thousand yards; otherwise the batteries are wrong. It appears to me that such a conclusion is legitimate, and whether this condition was reached by arguing about the range and then fixing the calibres of the guns to correspond or not does not matter; the fact is, I think, an interesting one to note as we pass. The old Constitution could penetrate a ship like herself at about 1,000 yards range. Those old ships of the Constitution class had a very heavy oak protection at the water line; in fact, they were relatively better protected than ships of to-day in many ways, and they could not penetrate each other's sides outside of a thousand yards, which I believe is the limit for ships of to-day also.

There appears to be no disposition to increase the caliber of the 12-inch turret guns forming the main battery of battleships, though there is considerable movement looking toward making these guns longer and heavier, and, consequently, imparting to their projectiles higher velocities than those now ruling. As respects the 6-inch guns forming the secondary batteries, naval opinion is entirely at variance. In this country there

are many who would substitute for the 6-inch guns 8-inch guns, and others who would substitute 7-inch guns, while in England the bulk of opinion, so far as discoverable from published accounts, is in favor of a 7.5-inch gun, whose projectile weighs 200 pounds. To this 200-pound projectile would be imparted a velocity of 3,000 or more feet, so that the 7.5-inch gun in England is a very large and very powerful weapon. The 7-inch gun proposed in this country and developed by our Navy Department, requires a 160-pound projectile, and to the projectile is imparted a velocity of 3,000 or more feet. There is some disposition on the continent of Europe to use lighter guns in turrets than those of 12-inch caliber, which will weigh from 45 to 50 tons; but in this country and in England the 12-inch gun seems to be pretty strongly held to. The secondary battery guns, however, as stated, are quite unsettled. Besides the 12-inch and 6-inch guns, protected, respectively, by 12-inch and 6-inch armor, battleships carry a large number of 3-inch 15-pdr. guns, having a high velocity and having very little protection. The large armored cruisers, which, in their elements of offence and defence, now approach battleships very closely, carry somewhat smaller guns than the latter and their armor is accordingly reduced. When we come to the smaller vessels of all classes, the protection afforded the batteries is usually very little, besides the gun shields, and the side and other armor, of course, is also materially reduced.

There are many reasons why the battleship, armed with 12-inch guns in her turrets and 8-inch guns on her broadsides, which is the favorite battery with those who would push the weight of battery very far, would have great advantages over battleships carrying only 6-inch guns in their secondary batteries; but, if the muzzle velocities of the 8-inch guns be equal to those of the 6-inch guns, and the total weights of battery are about the same, the number of 8-inch guns would be one-half that of the 6-inch and the rate of fire of the 8-inch, as compared with the 6-inch, will be very materially reduced—it is, perhaps, not too much to say that it will be reduced by one-half. The 6-inch gun can be man-handled—that is to say, the shot can be handled by one man and all operations performed with ease, without the use of mechanical apparatus. There are many devices for simplifying the handling of projectiles and getting them into the gun, but it must always remain that the weight of the 8-inch shot—now 250 pounds—will remain a serious impediment in the way of handling the gun rapidly.

Very little has been said about the numerous small caliber guns, such as 6 and 3 and 1-pounders. These are now usually either fully automatic or semi-automatic in character, and occupy an important field in certain

classes of naval warfare, such as the repulse of torpedo-boats, the protection of mine fields, and in landing operations. As regards a naval landing gun, it has been a rule to regard these in much the same way that the field gun on shore is regarded—that is, it is a question in which mobility is of great importance. Naval landing guns, however, must be materially lighter than the field gun of the Army, which weighs, on its carriage and without any ammunition, about 2,000 pounds. This is obviously too heavy to be drawn over rough roads by men, and the naval landing gun, itself weighing some 500 or 600 pounds, usually weighs, with its carriage complete, about 1,200 or 1,500 pounds, and is of approximately 3-inch bore. The 6 and 3 and 1-pounders are usually conveniently disposed on the rail of a ship and in her tops, and are suited only to the attack of unprotected parts. The 1-pounders and 3-pounders are also sometimes used as landing guns.

To summarize results it may be said that, in the past ten years, by the use of better mixtures, by improved heat treatment, and by very marked improvement in face hardening, the power of armor of a fixed thickness to resist penetration has been increased about 40 per cent., while its power to resist cracking has been increased in even greater degree. This has been taken advantage of in large measure in a reduction of the thickness of armor plate on a ship's side. The total weight of armor carried is not less than formerly, and indeed it is perhaps greater than it was, but it is more spread out, and, while the protection on the vital parts is as good and perhaps better than it was, there is now some weight in armor left to cover parts that formerly were without protection.

Gun steel also—and in speaking of gun steel we may include the steel for shafting as well—has been materially improved in the period under review. The shafting steel now going into ships is very much stronger than that which was used ten years ago. It is, indeed, perhaps quite 70 or 80 per cent. stronger than the former material. The advance in gun steel has not been so marked, because ten years ago the purchasers of gun steel wanted an elastic strength of about 40,000 pounds, whereas only 28,000 or 30,000 pounds was asked for in shafting. Now they both run 50,000 pounds, or even more. These very desirable and very material advances in quality have been rendered possible by the adoption of more satisfactory mixtures in the melting furnaces, and by more skilful heat treatment. They have not, however, been accomplished without much experimentation and the expenditure of a great deal of thought and money. The noted advance in the strength of gun steel has not been followed by a reduction in the thickness of gun's walls. Indeed the gun's thickness is rather greater perhaps

than it was, and the increased strength serves only as an additional guarantee against disaster.

When we come to what may be called the tactical aspects of guns and armor, we find also material changes. The quality of the gun steel and of the armor, as is stated above, has advanced by perhaps 40 per cent., and full advantage has been taken of this in design. The modifications in design which have come about have been numerous. The armor is more distributed. It has come to be recognized that unprotected guns have no chance, and that 6-inch and 12-inch guns should be protected, respectively, by 6-inch and 12-inch armor. It should be noted that this means, if battleships are to penetrate each other, that ships must approach within 1,000 yards range in combat. This brings us back to the fighting range of old wooden ships, very nearly, for they had to approach within about 600 or 700 yards; and, when you consider that a battleship's 6-inch guns must have normal impact to perforate the armor protecting the 6-inch guns of their opponents at 800 or 1,000 yards, it may be said that the fighting range of ships has not so very materially altered in the past hundred years. What is here said may perhaps be combated, but, as 2,200 feet velocity in a 6-inch projectile will not, even with normal impact, carry it through 6-inch armor, nor will 2,000 feet carry a 12-inch projectile through 12-inch armor under like circumstances, it follows, when the vast preponderance of oblique impacts is considered, that ships must fight within 1,000 yards.

Finally, in closing, I want to bring forward, in order to more definitely determine the coming range of naval combat, the fact that there is one of its ruling factors which remains constant for all time. This is, the accuracy and delicacy of the firer's eyes and nerves. If a man can appreciate four intervals in a second, and if the firing apparatus put into his hands is instantaneous in action, he can hit a ship 15 feet high, at a range of about 1,000 yards, when his own gun is swinging, in rolling, at an angular rate of one degree per second. This consideration, of course, remains fixed for all time. Another important consideration is what is usually called the "danger" space. With medium calibers, with 2,000 feet velocity, the danger space is 1,100 yards. That is to say, the gun is independent of the range, as the trajectory is so nearly flat at that distance. The new guns now coming along, having 3,000 feet velocity, have a greater danger space with existing targets, and are therefore independent of the range up to 1,300 or 1,400 yards.

Whether it is better to use a 7-inch or 7.5-inch gun on the same weight as the 50-caliber 6-inch gun—that is, about 18,000 pounds, in the secondary

battery of battleships, is one of the pressing questions of the day. If the 7-inch or 7.5-inch gun has its weight increased proportionately it will, of course, be better than the 6-inch, except that its rate of fire will be less from the weight of its shot, but, if we take the larger calibered gun and accept a somewhat lower velocity, while using the heavier and larger shot and the same total weight in the gun and carriage, it may well be argued that the larger gun is the better. On the other hand, high velocity gives always an independence of the range at greater distances, and gives always greater penetration. The designer of naval ships is therefore on the horns of a dilemma with regard to the battery of the ship, as indeed he must be with regard to pretty much all of her elements, and he must be guided by judgment and by a careful balancing of conditions, so that, upon the whole, and in the circumstances most likely to occur, the ship he produces will be able to give a good account of herself.

DISCUSSION.

THE CHAIRMAN:—Gentlemen, before proceeding to the discussion of Mr. Meigs' very complete paper we will have read a paper on a similar subject, "Recent Experiments in Attacking Armor with High Explosive Shells," and we can then discuss the two papers together, as our time is getting short. I will call on Captain Babbitt to read his paper.

THE SECRETARY:—I have just received a telegram from Captain Babbitt saying that, on account of the official tests of the Gathmann gun, he is unable to present his paper in person, and begs to express his regret at his unavoidable absence, and his sincere appreciation of the compliment of being invited to contribute a paper. I might say, in passing, that the résumé which Captain Babbitt has given is an exceedingly interesting one, and deals with a subject of the greatest importance, and I think the data submitted will be a great surprise to many of us, even those who have followed the subject quite closely.

The Secretary then read Captain Babbitt's paper.

RECENT EXPERIMENTS IN ATTACKING ARMOR WITH HIGH EXPLOSIVE SHELL.

By E. B. Babbitt, Captain of Ordnance, U. S. A.

[Read at the ninth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 14 and 15, 1901.]

The struggle between the armor and the projectile is drawing to a close. The limit of flotation has practically settled the thickness of armor to be carried, and the manufacturers now bend their energies, with remarkable success it is true, to *improving* the armor.

With all this, however, the projectile is in the lead, and has now brought to its assistance the valuable aid of high explosives.

The Ordnance Department of the Army has been experimenting for many years to secure a high explosive for a shell filler, and with varying success. In the reports of the Ordnance Board conducting these tests in 1883, 1884, and 1885, they usually concluded with "burst at the muzzle" or "destroyed the gun." Naturally, then, the problem, as it presented itself to them, was to secure an explosive that could safely be fired from a high-powered gun. As we approach it now, that is really but a corrollary to the condition that it will not explode on impact on armor, as the shock in the latter case is so much greater.

When the Board of Ordnance and Fortification was organized, the tests with high explosives took a wider range, due partly to the larger fund at its disposal and partly to the increased activity of the manufacturers for a time.

The subcommittee of this Board considered at length this subject and reported:—

It appears that the following points are to be investigated:—

1st. Which is the best explosive for use in shells.

2d. Is an explosion of the first order, or detonation, assured with the different methods of loading proposed by the inventors to prevent premature explosion.

3d. Will the explosive stand the friction due to rifled motion when loaded as proposed.

4th. Will the explosive stand the shock of firing when loaded as proposed by the different inventors.

5th. Will the igniting mechanism stand the shock of firing and the rifled motion without premature action.

6th. Will the igniting mechanism act as desired.

7th. Will the shell, without the igniting mechanism, explode on impact with water, with earth, with stone, with wood, or light iron, and with iron armor.

Considering these it appears :—

That the first is the summation of those that follow.

The second, in its investigation, led to extensive sub-terra experiments which, while instructive, really led the investigators from the true scope, namely, to get an explosive that could be used in a shell.

The third condition was quite a bugaboo, and comes to us even to-day. I have my doubts as to its importance.

In a moment of mental aberration, I devised a fuse depending for its action upon a plunger within remaining at rest while the envelope took up the rotation of the projectile. I had no doubt about this feature, but, wishing to investigate another, made such a plunger, supported, by the way, by an axial shaft, thus reducing the frictional moment, and fired it from a field gun into a sand butt. When recovered and opened—behold! the plunger had not moved in its housing. Thus ended a threatened revolution in the fuses of the world.

Of the remaining conditions, those regarding the fuse or firing mechanism are important, since in the later development a successful explosive waited on a satisfactory fuse.

At the time of this report, 1891, there were before the Board Explosive Gelatine, Guncotton, Emmensite, Schwalin Mixture, and Americanite. Of these Guncotton alone is now considered as a shell filler, and then only wet and with a matrix.

Between 1892 and 1896, there were tested in addition :—

Terrorite (N. G. and Nitro Ethyl and Nitromethyl).

Rackorock—a Spengle safety mixture.

Ammonite—90 per cent. Ammonium Nitrate and 10 per cent. Mono-Nitro-Naphthalene.

Jovite—87 NH_4NO_3 and Nitro-Phenols and Nitro-Naphthalene. None met with success. Between 1896 and 1898 the Ordnance Department experimented largely with guncotton, in various forms and degrees of moisture,

and met with considerable success, thanks to a detonating fuse devised by Lieutenant, now Captain W. S. Peirce, Ordnance Department.

In 1898 Thorite began its brilliant but short career.

The sensitiveness of most of the high explosives brought forth during this period many novel ideas for firing them with safety, the most notable being the famous pneumatic dynamite guns, three of which were mounted on the Vesuvius, and were used at Santiago with so little effect. As late as August 23, 1899, a charge of 129½ pounds explosive gelatine was fired from a 12-inch service rifle using an Isham diaphragm shell. The explosion of this mass against a 12-inch plate failed, on impact, to rupture the plate, but moved it bodily. More damage was done to a similar plate with a service armor piercing shot, uncharged.

With the reorganization of the Ordnance Board in 1899 was introduced the present system of investigation, which, in its workings, has brought about conclusive results, and has, it is thought, given a most satisfactory explosive and fuse; in other words, a most powerful weapon for attack, superior to any possessed abroad of which we have any knowledge.

I can do no better than quote the following reply of the Ordnance Board to a letter from the Chief of Ordnance asking about the status of the question of high explosives for shell.

"From the nature of the subject it is evident, and experience has amply demonstrated the fact, that the selection of a high explosive shell which shall be thoroughly efficient and yet safe beyond any reasonable doubt to store, handle, and fire under all conditions of service, presents a most difficult problem which offers chances of failure at every step and can be considered solved only when these requirements are satisfactorily established by exhaustive tests. The destructiveness of the explosive makes it a constant menace to the men and material using it, and a perfected projectile involves not only the explosive but the fuse and the method of constructing the shell also."

A satisfactory high explosive for shell should fulfill the following requirements:—

SAFETY AND INSENSITIVENESS.

1st. Should be reasonably safe in manufacture and free from very injurious effects upon the operatives.

2d. To warrant further test, must show a relatively safe degree of insensitiveness in the impact testing apparatus used by the Department.

3d. Must withstand the maximum shock of discharge under repeated firings in the shells for which it is intended.

4th. Must withstand the shock of impact when fired in unfused shells, as follows:—

a. Field shell. Maximum velocity against 3 inches of oak timber backed by sand.

b. Siege shell. Against seasoned concrete thicker than the shell will penetrate, with remaining velocity for full charge at 500 yards range.

c. Armor-piercing shell. Against a 7-inch tempered steel test plate in a 12-inch A. P. shell with a velocity just sufficient to penetrate.

Detonation and Strength.

5th. Must be uniformly and completely detonated with the service detonating fuse.

6th. Should possess the greatest strength compatible with other necessary requirements.

Stability.

7th. Must not decompose when hermetically sealed and subjected to a temperature of 120 degrees F. for one week.

8th. Should be preferably non-hygroscopic, and must not have its facility for detonation affected by moisture that can be absorbed under ordinary atmospheric exposure necessary in handling.

9th. Must not attack ordinary metals used in projectiles and fuses to an extent that cannot be prevented by simple means.

10th. Must not deteriorate or undergo chemical change in storage.

General Conditions.

Loading.—This must not be attended with unusual danger, and should not require exceptional skill or tedious methods. It is very desirable that the explosive be capable of compact loading either by melting or compressing into solid blocks.

Supply.—Should be possible to procure the explosive quickly and in quantity in this country at a reasonable cost.

To carry out investigations covering all these conditions involved constructing a comprehensive scheme of test and the expenditure of much money. The object being to get the explosive through armor, the final test would naturally be a practical trial under service conditions. These costing from $3,000 to $5,000 per round could be very few and means had to be devised for an early elimination of those doomed to failure.

We therefore first determined :—

1st. The relative force (by calculation) for actual density of loading in shell.
2d. Specific gravity.
3d. Density of loading.
4th. Charge contained in 100 cubic inches.
5th. Approximate cost of charge.
6th. Ease of supply.
7th. Method of loading.
8th. Safety in manufacturing.
9th. Stability in heat test.
10th. Non-hygroscopic.
12th. Non-active on metals.

If the above were satisfactory the explosive was tested in an impact testing machine devised by Captain Dunn, a member of the Board. In this is provided a shallow cup in an anvil capable of holding a few grains (by weight) of the explosive in such a manner as to prevent lateral expansion.

By noting the height of fall of a known weight, the relative insensitiveness is determined. It is of interest to know that, in every case, subsequent armor tests have borne out the conclusions reached with this little machine. Having once determined a limiting drop, below which an explosive would fail in a gun, it is easy to reject, without firing, all those below that limit, and indeed select only those approaching a higher drop limit that would indicate insensitiveness to impact.

A projectile acquires its velocity in passing over the entire length of the bore of a gun, but, if stopped by an armor plate, is arrested in a very small fraction of the first distance. It is, therefore, evident that the shock of impact is the severer test, and it is here that the tests become most interesting.

We have no means of determining the law of retardation in a plate, so for comparison are forced to assume that it is constant over the path.

Under this assumption, considering the length of column of the explosive in each service shell, and the armor against which it is tested, it can be

shown that the shock to the explosive in a 6-pdr. shell against a 3-inch nickel steel plate is greater than in any of the service tests. For this reason, and on account of the relative inexpensiveness, the 6-pdr. R. F. gun was selected for the preliminary tests against armor. Standard plates 1-inch, 1½-inch, 2-inch, and 3-inch, are used and firings made at each in order, till either the explosive bursts on impact, or successfully resists the highest plate.

The results of the first series of tests following this method are as follows :—

Summary of Tests of 6-Pdr. Steel Shell Charged with High Explosives and Fired (Unfused) Against Steel Plates of Varying Thickness.

Plates.—Braced against 6-inch oak backing (110 feet from gun) with intervening clear space of 20 feet to stand butt in rear—so placed in order to observe action of shell after penetrating target, and to collect fired shell or their fragments from the butt. The following plates were used :—

Carnegie tempered (nickel steel) plates, 36 x 36 inches.	1, 1.5 and 2 inches thick.
Fragments of mild steel plate.	3 inches thick.
Carnegie tempered (nickel steel) plate 38 x 40 inches.	3 inches thick.

Projectiles.—6-pdr. A. P. shell, steel, for which the acceptance test requires penetration of a three-inch mild steel plate, without fracture, base fuse dismantled and brass fuse stock cut off to be flush with the rear end of shell cavity when screwed home; hollow in stock filled with plaster of Paris. The projectiles were filled, flush with the inner end of fuse hole through base, with the following explosives :—

1st. Black musket powder, filled loosely into shell cavity—average charge 4 ounces.

2d. Rendrock Co.'s No. 400 explosive, heated to pasty condition, and tamped with wood stick in shell cavity—average charge 9.08 ounces.

3d. Picric acid, melted and poured into shell cavity—average charge 6.41 ounces.

4th. Guncotton pellets containing 15 per cent. moisture, loaded with matrix formed of 60 parts paraffine, 25 parts resin, and 15 parts beeswax—average charge 2.025 ounces.

5th. 10 per cent. Maximite, melted and poured into shell cavity—average charge 5.85 ounces.

6th. 25 per cent. Maximite, melted and poured into shell cavity—average charge 6.075 ounces.

7th. Explosive *D*, in powder, tamped as well as possible with wood stick through fuse hole—average charge 4.375 ounces.

Shell charge.	No. of rounds.	Plate (thickness), inches.	Action of shell.
Sand	2	1.5 T.	Penetrated target. Recovered whole.
Black powder	1	1.0 T.	Penetrated target and burst 4 feet in rear.
	1	1.5 T.	Penetrated target and burst close behind.
	1	2.0 T.	Penetrated target and burst 3 feet in rear.
No. 400 explosive	2	1.0 T.	Penetrated target and burst 30 inches in rear.
	1	1.5 T.	Penetrated target and burst close in rear.
Picric acid	2	1.0 T.	Penetrated target. Both had plug blown out between target and butt. Low order of explosion.
	2	1.5 T.	Both penetrated plate. One deflected to sea and lost; one exploded violently in backing.
	1	2.0 T.	Exploded in plate. Point passed through target and base blown to rear over gun shelter.
Guncotton	1	1.5 T.	Penetrated target and recovered whole. Charge slightly set forward at rear.
	1	2.0 T.	Penetrated target and exploded in butt.
10% maximite	1	2.0 T.	Penetrated target and entered butt apparently whole. Not recovered.
	2	3.0 M.	One penetrated target and recovered whole. Charge not disturbed. One stuck in plate with base imbedded 1.5 inches. Charge slightly set forward at rear.
25% maximite	1	1.5 T.	Penetrated target and recovered whole. Charge not disturbed.
	1	2.0 T.	Same, except charge slightly set forward at rear.
	2	3.0 M.	One penetrated target and recovered whole. Charge slightly disturbed at rear. One stuck in plate, head projecting about one-half through, not exploded.
Explosive "D"	2	2.0 T.	Penetrated target and recovered whole. Charge slightly disturbed at rear.
	2	3.0 M.	One same as preceding. One stuck in plate and broken in two across middle section; explosive not ignited.

(T., tempered; M., mild steel plate.)

It is seen that the black powder, No. 400 explosive, and picric acid, all failed to penetrate the 1-inch tempered steel plate without explosion; some superiority for picric acid is only shown in that the explosion was less violent than with the other two. The 1.5-inch plate produced a violent explosion of picric acid as well as of black powder and No. 400 explosive. Guncotton passed the 1.5-inch plate test but failed with the 2-inch plate.

The two grades of maximite and explosive "D" passed the 2-inch plate test and also the test with 3-inch mild steel to which they were subjected. A 3-inch tempered steel plate was then mounted to give an extreme test for these explosives. The powder charge was increased to give a striking velocity of 1,920 f/s. with which four rounds were fired, and also two rounds fired with a charge giving a striking velocity of 1,970 f/s., viz:—

Shell charge.	No. of rounds.	Plate (thickness), inches.	Action of shell.
Velocity 1,920 f/s.			
10% maximite.....	1	3 T.	Struck squarely, and point passed 4½ inches beyond front face of plate. Shell broken and fragments rebounded, leaving clean hole. Explosive ignited, as shown by cloud of smoke.
25% maximite	1	3 T.	Point penetrated about 4.25 inches. Shell rebounded 110 feet, badly set up. No explosion.
Explosive "D".....	2	3 T.	One with same penetration as preceding, and one striking somewhat obliquely, penetrated about 3 inches; both broken. Explosive formed cloud of powder; not ignited.
Velocity 1,970 f/s.			
25% maximite..	1	3 T.	Badly upset and broken on plate. Cup-shaped hole about 1.5 inches deep. Explosive ignited, as shown by cloud of smoke.
Explosive "D".....	1	3 T.	Same as preceding, except hole 1.7 inches deep. Explosive formed cloud of powder; not ignited.

These tests led to the selection of two explosives, namely, maximite and explosive "D," as worthy of final test.

Owing to the impossibility of controlling the explosives in an ordinary explosive chamber, fragmentation tests are made by burying the larger shell, fully charged and fused, about ten feet in sand and firing the fuse electrically.

Fragments are then dug out, weighed and counted. Careful search is made for any unexploded material.

The following extract from these fragmentation tests, gives an idea of the results obtained:—

1. 12-inch A. P. Shell, Maximite, buried 10 feet in sand. Frankford Arsenal A. P. Detonating fuse.

	lbs.	oz.
Weight of empty shell and fuse	960	4
" " explosive charge	50	12
	1011	0
Weight of metal recovered	827	7
Number of fragments counted	3690	
Number of fragments small pieces estimated	2718	

2. 12-inch A. P. Shell charged with Explosive "D," buried 12 feet in sand. Frankford Arsenal Detonating A. P. fuse.

Weight of empty shell and fuse	951– 4
" " explosive charge	57–12
	1009– 0
Weight of metal recovered	815
Number of fragments	850

The first result quoted above is much above the average and has not been duplicated. Indeed there is a limit to the size of useful fragments. A large number of fair-sized fragments driven with high velocity throughout the interior of a vessel would cause more damage than a much larger number of small fragments.

Such a statement is always subject to modifications; if, for example, a shell should burst in a crowded part of the vessel as was the case in the turret of the Japanese flagship at the battle of the Yalu, the more violent explosion would be more disastrous.

Both explosives having met all requirements up to this point there remained the final tests against armor.

Twelve-inch armor piercing shot having sufficient velocity for penetration, are required to pass through a 12-inch face-hardened plate before the lot represented by the projectile is accepted. This was, therefore, adopted as the final test for the explosives, namely, that they must withstand the shock of impact of such a projectile on such a plate. It was led up to by firing first a 5-inch shell through a 3-inch tempered plate. Next, a 12-inch A. P.

shell through a 7-inch tempered plate, and finally a 12-inch A. P. shot (capped) through a 12-inch harveyized plate.

All the above were unfused, and, in every case, the projectiles were recovered and, later, fragmented to increase our data on this subject.

It should be stated that all armor plates are supported by heavy backings of oak strongly braced, and in rear of this is a large mound of sand in which the projectile finally comes to rest. At last the final test was reached.

Imagine the tension as, from a safe distance, we stood with eyes to our glasses anxiously watching the distant field. The target, a 5¾-inch tempered steel plate, well backed with oak, defiantly faces the long slender 12-inch rifle. The red flag waves from the firing bomb proof. It is answered. The flag falls, a bank of smoke from the gun, a flash of light at the plate, a dense, black, forboding mass of smoke interspersed with flying timbers and bits of plate, two mighty roars in quick succession, a sigh of relief and satisfaction from the observers, and, for the first time, a 12-inch armor piercing shell, loaded and fused, has passed through heavy armor. While, later, heavier plates were used and the results therefore more satisfactory, the first success ever stands out most vividly in the minds of the experimenters.

The official record of this test reads as follows:—

No. 7. 12-inch A. P. shell charged with explosive "D," 58½ pounds.
Weight of charged shell complete 1,021 pounds.

Fired March 27, 1901, with Frankford Arsenal Detonating A. P. fuse complete against 5¾-inch tempered steel plate. Striking velocity 1,475 f/s. Shell penetrated plate and detonated immediately in rear. Plate broken and carried forward with numerous fragments of shell.

Tests against 7-inch steel plates and 12-inch face-hardened plates followed with great success. Finally, encouraged by the successful results, a 12-inch A. P. shell (the test of which is a 7-inch steel plate) was fired against a 12-inch face-hardened plate. The record reads:—

No. 11. 12-inch A. P. shell charged with explosive "D," 58.6 pounds.
Weight of charged shell complete 1010 pounds.

Fired May 17, 1901, with Frankford Arsenal Detonating A. P. fuse complete, against a piece of 12-inch face-hardened steel plate. Pressure in gun 29,900 pounds per square inch. Velocity about 1,875 f/s.

Shell detonated in plate and completely demolished plate and backing—all being carried forward and swept away. Fragments of plate were thrown to a distance of 200 to 300 feet, and gave evidence that the plate was penetrated by the shell.

In summing up the final tests we find:—

	Explosive "D."	Maximite.
Shell fragmented by detonating fuse	8	11
Shell fired from gun unfused	13	13
Shell fired from gun fused	13	10
	34	34
Results entirely satisfactory with respect to explosive,	34	33
Premature action of explosive, cases afterward remedied	—	1
Fuses acted feebly—remedied	3	1
Fuses failed	1	—

Before closing, it is necessary to mention a most important factor in this success, namely, the fuse. But, of this, little can be said. It is the secret of all our triumphs, and as such is being carefully guarded. It, like the explosive, passed through various stages. It was tested separately for its power to detonate the shell charges; its power to resist shock of discharge and impact; and, finally, under service conditions. It had its failures in the early stages, but, thanks to the skill and energy of Captain Beverly W. Dunn, Ordnance Department, U. S. A., won out in the end.

One of the most serious questions connected with an armor-piercing fuse is its delay action. Introducing, as it does, the time element, it is evident that an interval suited to light armor would not be satisfactory for heavy when the time of penetration is greater. As by far the greater part of a ship's protection is relatively light armor, a delay sufficient to cause detonation a few feet after penetration seems best. This gives us a burst when a shot is still in the 12-inch plate, but has penetrated.

I think I am safe in saying that the results just enumerated are unique, and so far surpass those previously obtained as not to admit of comparison with them.

APPENDIX.

[Submitted by author after the meeting.]

It was the original intention of the writer to present to the Society a paper on the Gathmann tests. These, however, were, for various causes, so long delayed as to preclude submitting anything at the regular meeting. The following notes are therefore added as a fitting ending to the paper actually read.

The so-called Gathmann tests were conducted in accordance with a specific appropriation of Congress, whereby a gun of 18 inches caliber, firing a torpedo projectile, charged with about 500 pounds of wet guncotton, was to be compared with a service 12-inch breech-loading rifle, firing service projectiles charged and fused for war conditions.

The comparison was to be made by the results obtained on firing against armor representing a section of the battleship Iowa. For this purpose two identical targets were erected in front of the competing guns.

These targets consisted of—

1. A Carnegie-Krupp face-hardened steel armor plate 16′ x 7½′ x 11½″ thick.
2. A layer of oak.
3. The steel cellular structure.
4. Oak backing supported by sand 26 feet deep.

Total weight, without sand, about 50 tons.

The test was virtually one between two rival systems.

The large capacity, low velocity, torpedo shell of the Gathmann system was intended to burst on impact and was expected to demolish all in front of it by the explosion of an enormous mass of explosive.

The small capacity, high velocity, thick wall armor-piercing shot of the service type was intended to penetrate the plate and then bursting destroy the structure within.

Military men generally agreed that the surface explosion, as proposed by the Gathmann tests, would be ineffective.

The fundamental principle of armor protection should be borne in mind

in considering this question. That is, the armor is expected to serve as a protection against but one shot of large caliber, it being assumed (and justly) that a second shot cannot be delivered at the same point.

Therefore, a comparison of the two systems should be made on the results of the first shot only.

So sure were the advocates of the Gathmann system that the act of Congress allowed "three shots from the Gathmann gun and ten from the service rifle."

The firing tests began November 15, and were concluded November 18.

It was intended that the striking velocity of each projectile would be that it would have at a range of 3,500 yards, if fired with full velocity.

Owing to the irregularities in the behavior of smokeless powders with the small charges required to simulate this range, the actual muzzle velocities obtained varied from those calculated.

Taken in the order in which fired, the results were as follows:

Round No. 1. Eighteen-inch Gathmann torpedo shell.

Weight empty	1,332 lbs.
Weight of wet guncotton	497 "
Weight of fuse	5 "
	1,834 "
Calculated velocity	1,650 f. s.
Actual velocity, muzzle	1,670 "
Pressure	11,900 lbs.
Striking energy, foot tons	35,000

The shell struck the plate in the center and exploded violently, forming an indentation in the plate about ¾" deep and about 6" in diameter.

The whole structure was moved about 4" to the rear, and the cellular structure buckled.

The plate was pratically unharmed. It was pushed 5½" to left of its original position.

All "x" plates behind the framing proper buckled 4" or 5" from a straight line.

Armor bolts found intact.

Upper edge of skin plates behind backing bent backward ⅝".

Deck on top of armor displaced to the rear about 1½".

All top bolts securing deck sheared off.

There were five uprights entirely uninjured.

Decking on top of armor driven backward from ½" to 1¼".

The plate was found dished 1½" on its major axis, and ½" on its minor axis.

Round No. 2. Eighteen-inch Gathmann torpedo shell.

Weight empty	1,327	lbs.
Weight of wet guncotton	502	"
Weight of fuse	5	"
Weight of added cotton and water	5	"
	1,839	"
Calculated velocity	1,650	f. s.
Actual velocity, muzzle	1,670	"
Pressure	12,600	lbs.
Striking energy, foot tons	35,000	

Aimed at center of right half of plate, as left by preceding shot.

The shell exploded violently on striking the plate, moved it 13" to the right. The plate remained intact, and a few very small pieces which were scaled off at the point of impact did not weigh over 2 pounds.

The point of the shell was discovered about 3 feet in front of plate, flattened out.

The right-hand edge of plate was moved 2' 9" to the rear; left side remained unchanged.

The dish in plate at point of impact horizontally measured 2" deep, and at center of plate 2¾".

No vertical dish at point of impact was found. The 3" steel plate in front of armor was cracked, extending ⅔ of its width. The plate was thrown 2' to the left of its original position.

The front edge of top deck plate over the point of impact was raised 4" above where it had been sheared at the first shot.

Round No. 5. Eighteen-inch Gathmann torpedo shell.

Weight empty	1,321	lbs.
Weight of wet guncotton	509	"
Weight of fuse	5	"
Weight of added cotton and water	5	"
	1,840	"

Calculated velocity	.....
Actual velocity, muzzle	1,990 f. s.
Pressure	18,500 lbs.
Striking energy, foot tons	50,509

Aimed at center of left half of plate.

The shell struck the point aimed at and detonated.

The plate was cracked vertically for its full height at the place struck in the preceding round.

The depth of the indentation in the plate did not exceed 1½" at any point, and its longest vertical diameter measured 18" and the horizontal 14".

The face-hardened surface of the plate was scaled off at the point of impact.

The plate was thrown 18" to the left.

The left-hand edge of plate was thrown backward about 15" under the top plate of coffer-dam.

The bottom left-hand side of plate was thrown back 4'. An examination of the structure was made with the following results:—

The whole structure in front of the braces was revolved to the left about 30°, and the top deck-plate turned 90° and thrown to the rear so its nearest edge was about 10' from the plate.

Round No. 2. Twelve-inch A. P. shot, capped.

Weight of shot and fuse	982	lbs.
Weight of explosive "D"	19 7/16	"
	1,001 7/16	"
Charge	237 2/16	lbs.
Calculated velocity	1,900	f. s.
Muzzle velocity	1,807	"
Pressure	21,100	lbs.
Striking energy, foot tons	22,520	

The shot penetrated the plate and detonated just in rear, tearing the cellular structure very badly and demolishing the backing.

All parts of projectile passed through the plate.

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The crater formed just in rear of the structure measured 15′ x 12′ x 6½ deep. Pieces of plate and projectile were thrown out of the crater and were recovered at distances of 150 and 200 feet in rear of the sand backing.

Round No. 4. Twelve-inch A. P. shot, capped.

Weight of shot and fuse	983	lbs.
Weight of maximite	23	"
	1,006	"
Charge	204 2/16	lbs.
Calculated velocity	1,900	f. s.
Muzzle velocity	1,812	"
Pressure	20,837	lbs.
Striking energy, foot tons	22,520	

The point of the shot penetrated the plate about 9½".

Shot detonated, cracking the plate.

The plate below the center of impact was set back 3" on the armor shelf, and the right-hand portion of plate raised 3 inches.

The plate was dished and pushed 8" to the rear at the point of impact.

The following cracks were made in the plate:—

Two through, starting from a common point and separated at a distance 1′ at edge of plate. One vertical extending from the point of impact to the upper and lower edges of plate, which measured 2" wide. One comparatively horizontal extending from the point of impact to the left-hand edge of plate. A crack also extended from crack No. 1 diagonally to the right and to the upper edge of plate.

The framing behind the right-hand edge of target was completely wrecked.

Diameter of hole at point of impact measured 15", and the rear portion was found separated from the front portion for a diameter of 3′.

All section bolts were carried away.

Two pieces of the plate, about 2 cubic feet each, were carried backward about 5 feet into the sand butt.

Round No. 6. Twelve-inch A. P. shell, capped.

Weight of shell and fuse	984 11/16	lbs.
Weight of explosive "D"	60 5/16	"
	1,045	"
Charge	237 1/8	lbs.
Calculated velocity	1,900	f. s.
Muzzle velocity	2,082	"
Pressure	28,550	lbs.
Striking energy, foot tons	29,800	

Upon this, the third impact, the front edge of the armor plate was torn and buckled downward. The lower left-hand corner in armor was broken into six large fragments, the largest about 4½′ square with the attached backing and skin plates torn loose from the structure and turned inwards, so that the left-hand edge of this fragment was in the plane of the original face of plate.

This shot resulted in the total destruction of the plate, frame, and coffer-dam. One large fragment, about 2′ x 2½′ x 11½″, weighing about 3,000 pounds, was hurled about 135 feet to the rear of the target, after passing through coffer-dam and about 15 feet of sand.

One piece of plate, about 15″ in diameter (punched through plate at point of impact), was carried through sand butt and dropped about 66′ in rear of plate.

One piece of plate, about a cubic foot in size, was found about 100′ in rear.

Point of projectile, weighing about 40 pounds, was recovered about 125′ in rear. In addition to those fragments specified, pieces constituting coffer-dam, and weighing about 50 pounds, were found from 25′ to 75′ in rear of plate; also smaller pieces of plate, coffer-dam and projectile, were found scattered about 100′ to right and left, and as far as 300′ to the rear.

Indications show that the shell exploded after penetrating about 5″ of the plate.

A good detonation.

This last shot is of special interest, since an armor piercing shell (the test for which is 7″ of nickel steel) was driven through 11½″ of Kruppized armor, the explosive charge undoubtedly largely aiding.

Considering the actual muzzle velocity obtained in Round 5 (with full charge of powder) it appears that the velocity of 1,670 f. s. obtained in the first round corresponded to a range of only 2,709 yards. Similarly the velocity of 1,807 f. s. in the second round corresponded to a range of 4,350 yards. It therefore appears that the 12-inch projectile would produce the results recorded in Round 2 at a battle range of 4,350 yards, while to produce the results in Round 1 the battle range for the Gathmann gun would be only 2,709 yards.

The following extract from the report of the joint board of officers of the Army and Navy, under whose general supervision these tests were made, summarizes the results obtained :—

"The act of Congress required that at least ten shots should be fired from the army rifle against one structure, but the annihilation of the structure by the three rounds made it impracticable to comply with this mandate.

"The results of these experiments give a fair illustration of the effect that can be produced by each system. The separate effect of each round and the accumulated effect of the three rounds from the army service rifle was, in each and every instance, superior to that from the Gathmann gun.

"After a careful consideration of the effect of the various impacts on the respective targets of the Gathmann gun and the 12-inch army service rifle, the board finds that none of the impacts from the Gathmann gun would have endangered a modern battleship; that the Gathmann system is not effective as a means of attacking armored vessels, and that any one of the shots from the 12-inch army service rifle would have wrought serious injury to a modern battleship as regards its buoyancy, the interior mechanism, the armament, and the personnel.

"It may be said in this connection that the destructive effect of the 12-inch army service rifle surpasses anything hitherto obtained from any gun, as far as this board has knowledge, or as the records show."

DISCUSSION.

THE CHAIRMAN :—The discussion of these papers is now in order. Mr. Fritz, would you care to say anything about the subject treated in these papers? I know the Society would like to hear from you.

MR. JOHN FRITZ, *Member*:—I am no speaker. I would rather make armor plate than talk about it.

LIEUTENANT COMMANDER WHITE, U. S. N.:—Mr. Chairman and gentlemen:—I wish to say that the practice quoted in this paper as having been made by the 6-inch gun is possible because the mechanism for handling the gun has been so perfected that it can be kept pointed at the target at all times and can be fired whenever loaded without waiting for the roll of the vessel as was formerly the practice; as the gun is loaded by hand, its movement does not interfere with this operation; the powder only has to be sent up from below, the projectiles being stowed in the casemate near the gun, which aids materially rapid loading. Accuracy of fire is, of course, the result of training, but the gun captain is supplied with a telescopic sight of such power that the flight of the shell may be followed till it strikes at medium ranges, also possible through the use of smokeless powder. The conditions of target practice may be ideal, but the results obtained give a fair basis for comparison. When we compare this practice with that made by guns of greater caliber, from 8 to 12 inch, the lighter gun has greatly the advantage, especially in the matter of accuracy of fire. The firing interval is increased by having to lay the gun in a fixed loading position, by the heavier weights to be loaded, and by the difficulty of finding the target with the heavier mass to be moved, both gun and turret.

The problem that is open to your mechanical ingenuity is where to reduce the weights, and how improve the mechanism for handling those that are indispensable. On account of the lack of accuracy of the fire of the heaviest guns I believe we are warranted in reducing the protection given to our 12-inch guns, replacing the turret by a shield.

NAVAL CONSTRUCTOR LINNARD:—There is one point in Captain Babbitt's paper which I think will materially affect the design of ships. If it is possible for armor piercing shells to be fired through seven or eight-inch armor and explode behind that armor, the casemate system is doomed, and all our guns have got to go into turrets, which certainly afford the best means of protection under those conditions.

THE CHAIRMAN:—Are there any other remarks? Mr. Meigs, have you anything to add?

MR. MEIGS:—No; nothing to add, thank you.

THE CHAIRMAN:—Gentlemen, the Chair, on account of the condition of his voice, is under the obligation of asking to be excused from reading a paper of his own, prepared in connection with Mr. Paulding, a junior member. Mr. Paulding does not care to face the Society, so the Secretary will kindly read the paper, "On Tidal Corrections."

THE SECRETARY:—Before reading the paper, I wish to move that a vote of thanks be extended to Mr. Meigs and Captain Babbitt for their exceedingly interesting papers. While general discussion has not usually been brought out by technical papers of this character, they are read over afterwards with the greatest interest, and, as the expression of expert opinion and the official record of important experiments, they are, when published in our proceedings, invaluable, and I think both gentlemen are entitled to a very hearty vote of thanks. I am sure it will be accorded.

The motion was put and unanimously carried.

The Secretary then read the paper.

SOME NOTES ON TIDAL CORRECTIONS.

By Edwin A. Stevens, Esq., Vice-President, and
C. P. Paulding, Esq., Junior.

[Read at the ninth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 14 and 15, 1901.]

In August last, through the kind interest of Engineer Lieutenant R. E. Hall, 2d Battalion, N. M. N. Y., and with the help of that officer and a party of his men, progressive trial runs were made on two days with the 30-foot steam cutter of the Brooklyn Naval Reserve. Data were carefully taken and recorded, and the results plotted. The curves are those marked with the dates of trial on Figure 1, Plate 51. The offsets were figured by the formula,

$$\text{speed in feet per minute} = \frac{R_a + R_w}{R_a t_w + R_w t_a} 6080.$$

The above formula was derived from that used in standardizing the screws of some of the earlier torpedo-boats as published in the Journal of the Society of Naval Engineers. As there given it is in the form—

$$X = \frac{R_w T_a + R_a T_w}{T_a + T_w}$$

in which X = number of revolutions to run one knot in still water.
R_w = number of revolutions on course with tide.
R_a = number of revolutions on course against tide.
T_w = time on course with tide.
T_a = time on course against tide.

It being more convenient for our purpose to express speed in feet per minute, the formula as used was algebraically deduced from the one just given.

The disagreement was so great as to apparently render the results valueless. These runs formed part of an extended series of trials, and as

every precaution had been taken to ensure accuracy, the outlook for any valuable result seemed gloomy. After verifying the length of the course, displacement and trim, and then checking every figure over again, the conclusion appeared to be that the error must be due either to faulty observations or to the tide. To eliminate the latter error it would be necessary to make at least four, preferably six, runs at each speed, which the results aimed at and the time available rendered impossible.

In considering the formula above mentioned it seemed possible to derive therefrom the data for a tide curve.

This formula rests on the assumption that the rate of change of tide and the advance of vessel per revolution are constant during each pair of runs.

Let N = flow of tide in feet per minute.

R_w and R_a = total revolutions to cover course with and against tide, respectively.

t_w and t_a = time in minutes with and against tide, respectively.

C = length of course in feet.

Then advance of vessel per revolution $= \frac{C - Nt_w}{R_w}$ or $\frac{C + Nt_a}{R_a}$.

Equating these values and solving for N,

we get $$N = C \frac{R_a - R_w}{R_a t_w + R_w t_a}.$$

This will give spots through which a tide curve can be drawn.

From this curve the tide for each run may be determined and the true speed thus obtained. The result of such a treatment is shown in the curves given in Figure 1. These curves so nearly approach each other that the difference may be due to errors of observation rather than to the inaccuracy of the assumptions on which the above formula is based, and the mean between them can be fairly taken as the true performance of the vessel under test.

The tide curves used in this calculation are shown in Figure 2, Plate 51.

In order to further test the formula an assumed tide curve was laid down as shown in Figure 3, Plate 52, and it was supposed that a vessel made four pairs of runs over a course of 6,080 feet at speeds of 400, 600, 800 and 1,000 feet per minute, with slip of the screw of 10, 11, 12, and 13 per cent., respectively. It was assumed that the trial was repeated by another vessel making

the same runs at about the same time but in the opposite direction. The first set of trials were called " a " and the second " w."

By using the tide curve the time of each run was ascertained and the total revolutions and R. P. M. were calculated, thus furnishing a record of the supposed progressive trials. These results are given in Table 1.

In Table 2 the data are treated by taking the mean of the apparent speeds of each pair of runs.

In Table 3 the formula $S = \frac{R_a + R_w}{R_a t_w + R_w t_a} 6080$ is applied.

In Table 4 the method above suggested was used to calculate the tide curves shown in Figure 3. The results obtained from Tables 2 and 3 are plotted in Figure 4, Plate 51, with a curve representing the assumed speed.

Tables 2 and 3 give practically the same results. The curves from Table 4 are omitted as they are almost identical with the curve for assumed speed.

The method suggested by Naval Constructor Taylor, U. S. N., of plotting the results of the runs against tide, and with the tide separately, and thence deducing a mean curve will, under the assumed conditions, give the same result as Table 2. In actual practice it has the great advantage that a variation in R. P. M. does not affect the result.

The difference between the curves on Figure 1 is at once explained by the fact that the first runs were made in opposite directions of the tide. It is evident that the errors of all methods are greatest at the lower speeds and may be either plus or minus.

It will be noticed that an error in the estimate of speed of tide entails but a small error in the speed of the ship except at very low speeds. For example, an error of two per cent. in estimating the speed of a 2-knot tide would mean an error of one-half of one per cent. at a speed of 8 knots and one per cent. at a speed of 4 knots.

The writer has never met the formula herein suggested, but in view of its very simple derivation, it seems probable that it has been used. Its application would offer no great advantage where the tide could be determined by actual measurement or when it was possible to get four runs at the same speed.

Where, however, as in this case, the vessel could not be spared for more than a limited time, separate tide observations were not practicable, and data from low speed trials were wanted, the method gives decided help in obtaining accurate results.

It can also be used to correct the results of trials of four or six runs at about the same speed where, as in the writer's experience, it has been impossible to maintain the R. P. M. absolutely constant, but where the departure has not been great enough to appreciably affect the slip of the screw.

It has been suggested that tidal eddies might introduce conditions changing so rapidly and irregularly as to be beyond the scope of any such mathematical treatment as proposed. While such may be the fact in isolated cases the writer has applied this method to four sets of runs with satisfactory results and with about the expected variation from the results obtained from the mean of apparent speeds; the tide curves obtained in each case were "fair."

The results of these trials and of others now being made the author hopes to be able to present to this Society at some future date.

TABLE I.

RECORD FOR SUPPOSITITIOUS CASE.

Run.	Start.	Time finish.	Time on course.	Mean hr. of run.	Tide from curve.	Revolutions.		Slip.	True speed ft. per min.
						Total.	Per min.		
"a"	*h. m.*	*h. m.*	*m.*	*h. m.*				%	
1a*	1-30	1-50.27	20.27	1-40	100	901			
2w†	2-20	2-30.76	10.76	2-25	165	478	44.4	10	400
3a	3-	3-15.91	15.91	3-08	218	1.073			
4w	3-40	3-47.15	7.15	3-44	250	482	67.4	11	600
5a	4-05	4-16.47	11.47	4-11	270	1.041			
6w	4-40	4-45.63	5.63	4-43	280	511	90.91	12	800
7a	4-55	5-03.45	8.45	5-	280	970			
8w	5-20	5-24.77	4.77	5-22	274	548	115	13	1,000
"w"									
1w	1-30	1-42.34	12.34	1-36	93	548			
+2a	2-10	2-35.76	25.76	2-23	164	1.145	44.4	10	400
+3w	3-	3-07.40	7.47	3-04	214	503			
+4a	3-30	3-47.27	17.27	3-39	248	1.164	67.4	11	600
5w	4-05	4-10.70	5.70	4-08	267	518			
6a	4-30	4-41.62	11.62	4-36	277	1.057	90.91	12	800
7w	4-55	4-59.75	4.75	4-57	280	546			
8a	5-15	5-23.39	8.39	5-19	275	965	115	13	1,000

* *a* = against. † *w* = with.

TABLE 2.

ANALYSIS—METHOD 1, BY MEANS OF RUNS.

Run.	Time on course.	App. speed.	Means.	True speed.	Error.
"*a*"	*minutes.*	*ft. per min.*			%
1a	20.27	300			
2w	10.76	565	433	400	+8.25
3a	15.91	382			
4w	7.15	850	616	600	+2.67
5a	11.47	530			
6w	5.63	1,080	805	800	+0.62
7a	8.45	720			
8w	4.77	1,275	997	1,000	−0.3
"*w*"					
1w	12.34	493			
2a	25.76+	236	364	400	−9.
3w	7.47+	814			
4a	17.27+	352	583	600	−2.83
5a	5.70	1,067			
6a	11.62+	523	795	800	−0.62
7w	4.75	1,280			
8a	8.39	725	1,002	1,000	+0.2

TABLE 3.

Method 2: Using formula $S = 6080 \frac{R_a + R_w}{R_w t_a + R_a t_w}$ when S = true speed in feet per minute.

Run.	Time on course.	Rev.	R_a+R_w.	$R_a t_w$.	$R_w t_a$.	$R_w t_a + R_a t_w$.	S.	True speed.	Error.
"a"	*min.*	*total.*							%
1a	20.27	901							
2w	10.76	478	1,379	9,700	9,700	19,400	433	400	+8.25
3a	15.91	1,073							
4w	7.15	482	1,555	7,680	7,670	15,350	616	600	+2.67
5a	11.47	1,041							
6w	5.63	511	1,552	5,860	5,860	11,720	804	800	+0.5
7a	8.45	970							
8w	4.77	548	1,518	4,620	4,630	9,250	997	1,000	−0.3
"w"									
1w	12.34	548							
2a	25.76	1,145	1,693	14,130	14,130	28,260	364	400	−9.
3w	7.47	503							
4a	17.27	1,164	1,667	8,700	8,700	17,400	583	600	−2.
5w	5.70	518							
6a	11.62	1.057	1,575	6,250	5,820	12,070	794	800	−0.75
7w	4.75	546							
8a	8.39	965	1,511	4,580	4,580	9,160	1,003	1,000	+0.3

TABLE 4.

Method 3: Formula $N = C \times \frac{R_a - R_w}{R_a t_w + R_{wa} +}$.

Run.	Time on course.	Rev.	$R_a - R_w$	$\frac{R_a t_w}{R_a t_w + R_w t_a}$	N.	Mean time of day.	Tide fr. curve.	Corr.	Corrected course.	S.	True speed.	Error.
"a"	*min.*	*total.*										%
1a	20.27	901					107	+2170	8,250	407	400	+1.75
2w	10.76	478	423	19,400	133	2-0	166	−1790	4,290	398	400	−0.5
3a	15.91	1073					217	+3450	9,530	600	600	
4w	7.15	482	591	15,350	234	3-24	252	−1800	4,280	598	600	−0.33
5a	11.47	1041					260	+3090	9,170	799	800	−0.12
6w	5.63	511	530	11,720	275	4-25	278	−1565	4,515	803	800	+0.37
7a	8.45	970					278	+2350	8,430	996	1000	−0.4
8w	4.77	548	422	9,250	277	5-10	274	−1310	4,770	1000	1000	
"w"												
1w	12.34	548					91	−1120	4,960	402	400	+0.5
2a	25.76	1145	597	28,260	129	2-03	157	+4040	10,120	393	400	−1.75
3w	7.47	503					209	−1560	4,520	603	600	+0.5
4a	17.27	1164	661	17,400	231	3-24	245	+4230	10,310	598	600	−0.33
5w	5.70	578					265	−1510	4,570	803	800	+0.37
6a	11.62	1057	539	12,070	272	4-23	276	+3210	9,290	799	800	−0.12
7w	4.75	546					278	−1320	4,760	1001	1000	+0.1
8a	8.39	965	419	9,160	278	5-09	275	+2310	8,390	1000	1000	

* See Table 3.

DISCUSSION.

THE CHAIRMAN:—Gentlemen, are there any remarks on this paper?

MR. C. P. PAULDING, *Junior*:—In this method of working up the progressive trial trip, the curve of the tide is derived by a method which is about equivalent to the ordinary method of mean speed, and this is used as a first approximation, and the increased accuracy due to the method is because you start with a first approximation and then go on from that and make a second approximation. The practical application of the method, as stated in the paper, was this:—On a 30-foot steam-cutter, a set of progressive cards was made on one Sunday, and, on the next Sunday, another set was made. On Plate 51, in the lower left-hand corner, are shown the two curves, one lying considerably above the other. As stated in the paper, there was no way of accounting for the different results, and this method was then applied, and the two curves that so closely approach each other in outline, are shown higher up on the same plate, Plate 51. This is rather an extreme case of progressive trial. The speeds are very low on account of the size of the boat—only 4½ knots at the lowest speed. It is difficult to get accurate results with such low speeds, and I think it is fairly good proof of the value of the method to observe the bringing into coincidence the results of two progressive trials which, at first sight, seem to differ so much. The remainder of the paper simply presents a theoretical case to show the theoretical accuracy of the different methods. I think that the practical illustration brings out the value of the methods a good deal better.

THE CHAIRMAN:—Any other remarks? If not, we will pass to the next paper, "Side Launch of Torpedo-Boats and Torpedo-Boat Destroyers," by Assistant Naval Constructor W. G. Groesbeck, U. S. N.

Mr. Groesbeck read his paper.

SIDE LAUNCH OF TORPEDO-BOATS AND TORPEDO-BOAT DESTROYERS.

By Assistant Naval Constructor W. G. Groesbeck, U. S. N., Member.

[Read at the ninth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 14 and 15, 1901.]

Although side launching on the Lakes is not an uncommon practice, and the members of this Society are doubtless familiar with the details thereof, the side launches described in this paper present some novel features; and while the necessity for launching torpedo craft sideways may not again occur, a record of what has been done may be of interest, if not of value, to the members of this Society.

This paper describes the launching of torpedo-boats Shubrick, Stockton, and Thornton, and of torpedo-boat destroyers Dale and Decatur, all of which were built by the William R. Trigg Company, at Richmond, Va.

Before proceeding with the subject-matter of this paper, it may be proper to remark that these vessels were not launched sideways from choice; on the contrary, the limited extent of land and water available at the time these vessels were laid down left the Trigg Company no alternative. The land at their disposal was a wedge-shaped piece of ground, bounded on two sides by the old James river and Kanawha canal and by the dock at the head of the same, and on the third side by the tracks of the Chesapeake and Ohio Railroad and of the Southern Railway. The canal is about 100 feet wide, and is normally about 15 feet deep, although this was increased by flooding during the launches to about 18 feet. The Shubrick's building slip occupied the apex of the wedge, and the Stockton and Thornton, and Dale and Decatur, were built in pairs in the order named. It may also be remarked here that the vessels were launched in the order in which they have been mentioned above. The Shubrick, Stockton, and Thornton are sister vessels, as are the Dale and Decatur.

It may be said that in general the methods used for the two classes, torpedo-boats and destroyers, were the same, with certain necessary variations. In order to make this article as brief as possible, only the launchings of the torpedo-boats have been described in full. For the destroyers men-

tion of the variations has been made. For the same reason, in the plans accompanying this description, reference to the Shubrick has been omitted, because the arrangement of the building slip and launching ways was exactly similar in her case to those used for the Stockton and Thornton. These were as shown on Plate 53.

Reference to Figure 1, Plate 53, will show very clearly the arrangement of piling used in the building slip, and a detailed description of the same is not considered necessary. The keels were laid horizontal and parallel to the edge of the canal.

The groundways were of 12 inch by 12 inch yellow pine, and were eight in number, the inclination being 2 inches to 1 foot. This, I believe, is about the inclination of the groundways used on the Lakes for side launches; but this is about the only point in which there is any resemblance between the methods adopted on the Lakes and for the vessels under consideration. Two points of radical difference appear. Reference to Plate 55 of Volume 7 of the Transactions of this Society, which the Secretary has kindly consented to republish for my benefit as Plate 58, will show clearly the two points in question. This plate shows a photograph of the steamship Maunaloa ready for launching.

It will be seen from this photograph that the groundways are placed about equal distances apart, and that the cradle is built up under the ship with the lower members or sliding ways at right angles to the groundways, and with their ends projecting some feet over their respective groundways. The sliding ways also appear to be free to move across the groundways, so that, with this method of launching, it would not make any difference if one end of the vessel were to start a little sooner than the other end. For the boats under discussion, the eight groundways were arranged in four groups of two each, as shown on Plate 53. The two members of each group were 15 feet apart, and the cradles were built up in the form of rigid rectangles, the sliding ways being parallel to and resting on the groundways throughout their length. They were kept from slipping off the groundways by shoes similar to those used in ordinary end-on launches. With this arrangement it was, of course, very important that both ends of the vessel should start at the same time, as, if one end were to gain a decided start over the other end, it might happen that the sliding ways would ride up over the groundways and slip off, or, as an alternative, the cradles might be broken up. So far as I can see, the method adopted by the Trigg Company possesses no decided advantage over the method used on the Lakes, and I believe that in future launches the Trigg Company propose to make

their practice conform to the Lake practice. It is proper, however, to remark that had the Trigg Company used the Lake method in the case of these boats, whose ratio of beam to length is about 1 to 10, the groundways would have had to be increased in number, and the expense of the launching would have been correspondingly increased. As will be seen later on, there was some little trouble in getting both ends of the vessels to start at the same time, but there never was enough difference between the times of the starting of the two ends to materially affect the success of the launch.

The second point of difference mentioned above is found in the length of the groundways and the relative positions of their ends with reference to the water-line. In the case of the Maunaloa the groundways will be seen ended on the water's edge and rounded off. The effect of this is that the vessel turns over slightly on her bilge in launching and *drops off* the ways, if I may be allowed to use the expression. Reference to Plate 53 will show that the groundways for the torpedo-boats were carried down well under the water. The boats were thus entirely water-borne before leaving the ways, and *floated off*. I do not know that carrying the ways down under the water was absolutely necessary. But as it saved some strain on the hull during launching, the precaution taken is considered to have been distinctly worth while; and as it was possible to lower the water in the canal sufficiently to do the entire work above water, the extra expense was not great.

Of the cradles themselves it seems to be necessary to say very little, as they are shown on Plate 53 in sufficient detail to enable them to be readily understood (Figures 2, 3, and 4). Nor do the wedging up and preparations for launching seem to call for any special comment, as they were carried on in exactly the same way as they would have been for an end-on launch. The releasing devices, of which there were four, one to each cradle, are shown in position in Figure 1, and in detail in Figure 5, Plate 53. An oak balk, "A," about 4½ inches by 3 inches, was firmly bolted to the sliding ways. The trigger "B," bore on its inner end against a 3-inch depression cut in the groundway and also against a 4-inch by 6-inch balk bolted thereto. The dog "C" was inserted between "A" and "D," as shown, and the whole was retained in place by the trigger rope. This apparatus was put in place just before launching, and as much strain as possible was set up in the trigger rope by means of a four-fold tackle, which was then belayed. After wedging up and removing the blocking, the vessel was held in place by the four trigger ropes. To launch it was only necessary to cut the four trigger ropes simultaneously. An ordinary 12-inch by 12-inch pine ram was ready on each groundway at the upper end of the cradle for use in starting if necessary.

Plate 54 shows the arrangement for the two destroyers. There were five sets of groundways, instead of the four used for the boats, and the scantlings used in the cradles were in general somewhat heavier. There were five cradles instead of four and five sets of triggers. Other minor differences will be noticed in the details given below, all of which were made necessary by the larger dimensions and launching weights of the destroyers.

TABLE OF LAUNCHING DATA.

	Shubrick.	Stockton.	Thornton.	Dale and Decatur.
Groundways	12″ x 12″ yellow pine			12″ x 12″ Y. P.
Groundway shoes	12″ x 1½″ oak			12″ x 1½″ oak.
Sliding ways	4¾″ x 9¾″ oak			5½″ x 12″ oak.
Sliding ways, length	16′			23′.
Inclination of ways	2″ to 1 foot			2″ in 1 foot.
Number of ways	8			10.
Number of cradles	4			5.
Number of triggers	4			5.
Bearing surface	103⅓ sq. ft.			230 sq. ft.
Launching weight	58 tons	64.62 tons	88.15 tons	193 tons.
Pressure per sq. ft.	00.56 tons	0.625 tons	0.853 tons	0.839 tons.
Draughts after launch:				
Forward	0′ 8½″	0′ 6″	1′ 10″	1′ 10½″.
Aft	6′ 0″	6′ 6″	6′ 3¾″	7′ ½″.
Mean	3′ 4¼″	3′ 6″	4′ ⅞″	4′ 5½″.
Length over all	175′			245′.
Beam	17′ 7½″			23′.
Depth	12′ 1⅛″			14′ 3″.
Draught	4′ 8″			6′ 6″.
Designed displacement	170 tons			420 tons.
Designed I. H. P.	3,000			8,000.
Boilers, Thornycroft	3			4, Daring type.
Total grate surface	137 sq. ft.			315 sq. ft.
Total heating surface	7,550 sq. ft.			17,768 sq. ft.
Steam pressure at boiler	250 lbs.			300 lbs.
Twin screw engines, 4 cylinder triple expansion.	Cylinders 14″ x 22″ x 25¼″ x 25¼″, stroke 18″.			Cyl. 20½″ x 32″ x 38″ x 38″, stroke 22″.
Speed, estimated	26 knots			28 knots.

In the above table the particulars as to the weights of the Shubrick, Stockton, and Thornton are detailed. For the Dale and Decatur the weights

are so nearly alike that it is not worth while to differentiate them. The figures given are for the Dale.

In all the launchings a preparation of one pound No. 2 Albany grease to five pounds of beef tallow was used. The grease was boiled three hours and then mixed with the melted tallow. After application to the ways the surface was covered with lard oil. In very warm weather the proportion of the mixture was altered by the addition of five pounds more of beef tallow, which was put in after the original stock mixture had been heated.

Plates Nos. 53 and 54 show the Thornton inboard of the Stockton, and the Decatur inboard of the Dale; the vessels being shown on these plates in their building positions. As it was not considered safe to launch them from a position so far from the water, it became necessary, after launching the Stockton and Dale, to lower the Thornton and Decatur to the positions previously occupied by their sister vessels. The means employed were similar in both cases, and are illustrated in Figure 6, Plate 53, and Figure 7, Plate 54. As the only difference between the operations on the Thornton and Decatur was in the number of jacks employed, there being one for each groundway, a description of the method used on the Thornton will be sufficient.

The groundways were bored throughout with sets of two holes each, for 1-inch bolts. Eight 12-inch by 12-inch by 4-foot blocks of yellow pine were prepared, and holes drilled in them to correspond with the holes in the groundways. The holes in the groundways were so arranged that the pine blocks could be shifted down about 24 inches at a time. There being one jack for each groundway, there were two on each cradle, and the positions of these two were so arranged that one was always about 12 inches in advance of the other. The jacks used were 8-ton screw jacks. The operation of lowering needs but little description. All the jacks being in place and set up, the vessel was wedged up and the blocks removed, as for launching. The jacks were then backed down until the four nearest the cradles could go no further. The lowering was stopped and the jacks removed; the blocks were then shifted back 24 inches, the vessel being held in place by the other four jacks. The jacks were then replaced and the lowering was continued until the other set of jacks had to be shifted back, and so on. The operation was slow, but entirely successful. The total traverse in the case of the Thornton was 30 feet, and in the case of the Decatur 32 feet.

Having now gotten all our vessels ready for launching, a few words as to the actual launching of the several ships will properly bring this article to a close. I regret that at the time of these launchings the preparation of

this article had not occurred to me, and I am therefore compelled to rely on my memory, as I took no notes. The record would have been, however, about as follows:

Shubrick, October 30, 1899; raining, warm. Vessel launched without shafting, rudder, or propellers. Rather slow in starting and rams had to be used. Entirely successful.

Stockton, December 27, 1899; clear and cold. Shafting, propellers, and rudder in place. Grease very hard and vessel was very slow in starting. Did not start until after continued applications of the rams. Stern started first and gained about 6 feet start of the bow. Forward cradles strained, and some of the fastenings started. There was no damage to the hull.

Thornton, May 15, 1900; clear and warm. Conditions the same as for the Stockton, except that about 23½ tons of pig iron was put in forward compartment to counterbalance the weight of shafting, etc., and to equalize and increase the pressure on the ways. Launch entirely successful. Rams unnecessary.

Dale, July 24, 1900; clear and very warm. About 20 tons of pig iron in forward compartment to equalize the pressure on the ways. Launch entirely successful. Rams unnecessary.

Decatur, September 26, 1900; clear and very warm. Conditions as to ballast same as for Dale. Launch was entirely successful, except that through a mistake of the axe man at the midship trigger, this trigger rope was not cut with the others. The vessel started immediately, however, without the use of the rams, and tore out the midship trigger device. The vessel started so quickly that the delay due to the midship trigger rope not being cut was not noticeable, and the fact was only discovered after the launch was over.

Plate 55 is a photograph of the Shubrick just before launching, and shows the arrangement of cradles and groundways used for the torpedo-boats. Plate 56 shows the Dale ready for launching, and plate 57 shows the Decatur just as she took the water. The three torpedo-boats launched had only sufficient impetus to carry them about 30 feet from the water's edge, while the two destroyers were carried about half way across the canal. No snubbing lines were used, although lines were fitted to hold the vessels from the banks of the canal after the launch was over and for convenience in handling.

DISCUSSION.

THE CHAIRMAN:—Are there any remarks on Mr. Groesbeck's paper.

MR. FORBES:—Mr. Chairman, this paper is most interesting, and it fully portrays a system of launching I believe thoroughly well understood and practiced on the Lakes. In this part of the world, it is something that is unusual, but from the complete data and diagrams presented by the author, there should be little trouble in following this method when the restrictions of location compel the use of side launching. I take pleasure in moving that a vote of thanks be extended to the writer of this paper, Mr. Groesbeck, for his very able contribution.

The motion was unanimously carried.

THE SECRETARY:—The following invitations have been received:

The American Society of Mechanical Engineers takes pleasure in extending to the Society of Naval Architects and Marine Engineers an invitation to attend the unveiling of the Fulton memorial in Trinity Churchyard, New York, on December 5, 1901. Formal invitations will be extended to those desiring to attend.

A cordial invitation has also been extended by the "Schiffbautechnische Gesellschaft" to attend their summer meeting in Dusseldorf, from June 2d to 5th, 1902. Programmes of this meeting will be issued when received.

THE CHAIRMAN:—Gentlemen, acknowledgments of the courtesy of the Society of Mechanical Engineers and the "Schiffbautechnische Gesellschaft" are in order.

MR. W. H. FLETCHER:—I make such a motion, sir.

THE CHAIRMAN:—It is moved and seconded that the thanks of this Society be extended to the American Society of Mechanical Engineers for their courteous invitation to participate in the unveiling of the memorial of Robert Fulton, and to the "Schiffbautechnische Gesellschaft" for their cordial invitation to attend their summer meeting in June, 1902.

The motion was carried.

THE SECRETARY:—We are under many obligations to the American Society of Mechanical Engineers for the use of their rooms. This privilege adds greatly to our comfort, and I move that we give them a vote of thanks for their courtesy.

The motion was carried.

THE CHAIRMAN:—Is there any other business to come before the Society? If not, it becomes my duty——

MR. W. H. FLETCHER:—Before we adjourn, Mr. Chairman, I think a vote of thanks is due to Colonel Stevens for conceiving the idea and offering the prize which produced such very valuable and interesting papers on "The Balancing of Marine Engines," and I therefore make such a motion.

THE CHAIRMAN (COLONEL STEVENS):—The Chair would inform the gentleman that he has no official knowledge of the identity of the gentleman who offered the prize, and must therefore declare the motion out of order.

Gentlemen, it becomes my duty to declare this meeting adjourned *sine die*.

Deaths, 1901.

NATHANIEL R. BENSON.

FRANCIS M. BUNCE, U. S. N.

HENRY W. CRAMP.

WILLIAM E. FITZGERALD.

To illustrate paper on "Trial of Speed between the Steamers City of Erie and Tashmoo," by Frank E. Kirby, Esq., Member of Council.

To illustrate paper on "Trial of Speed between the Steamers City of Erie and Tashmoo," by Frank E. Kirby, Esq., Member of Council.

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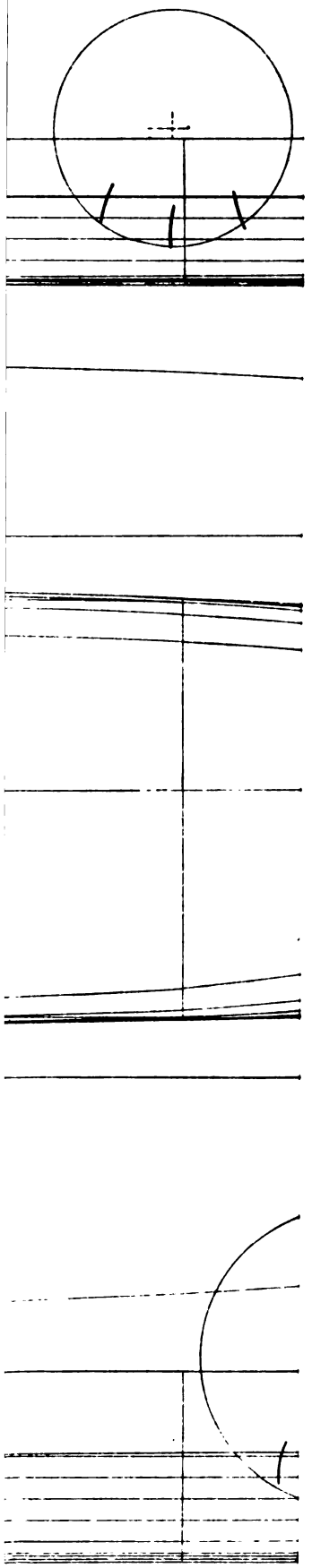
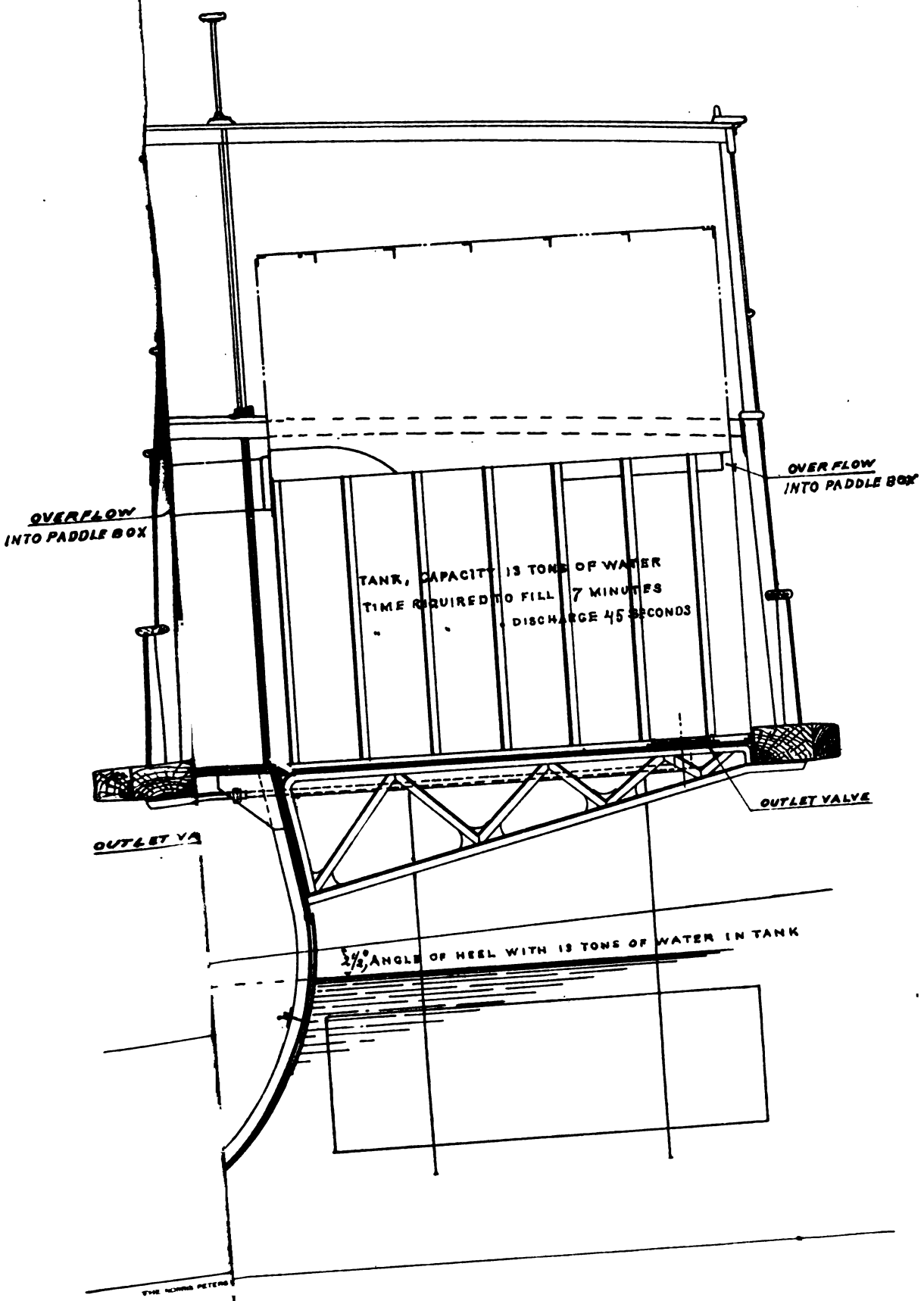
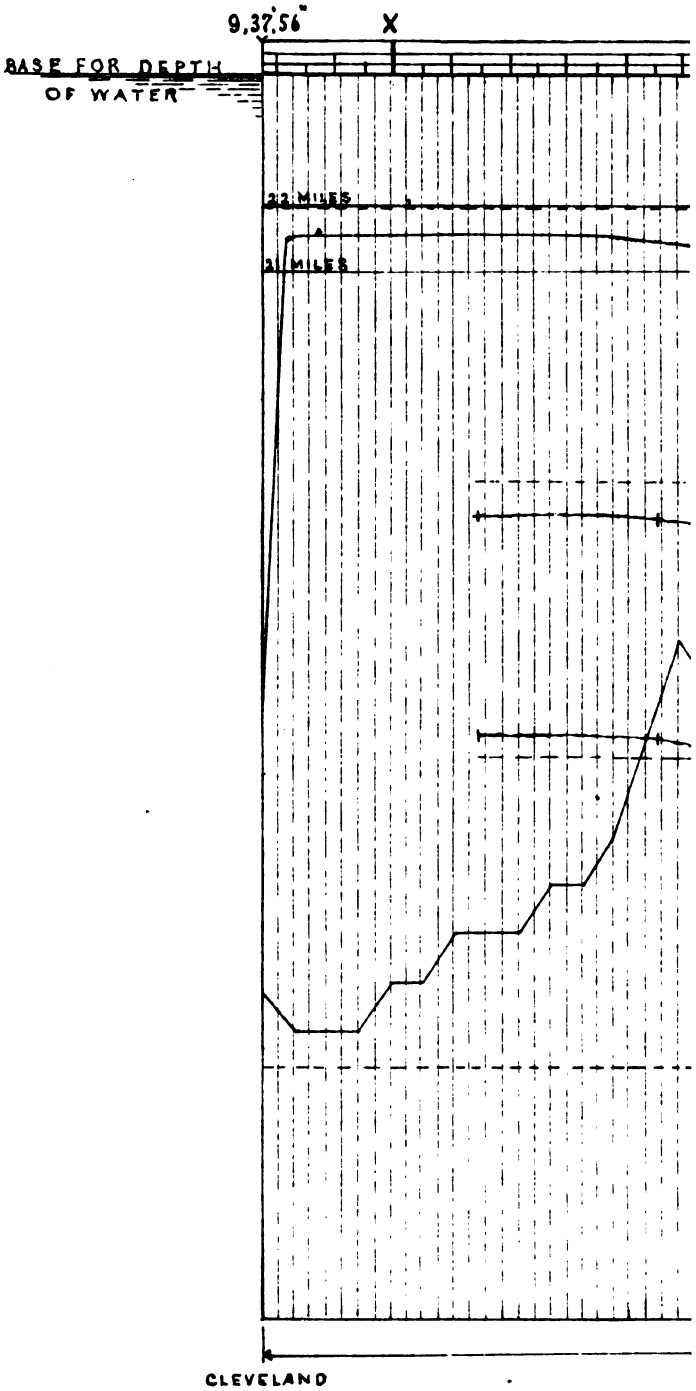
ber of Council.



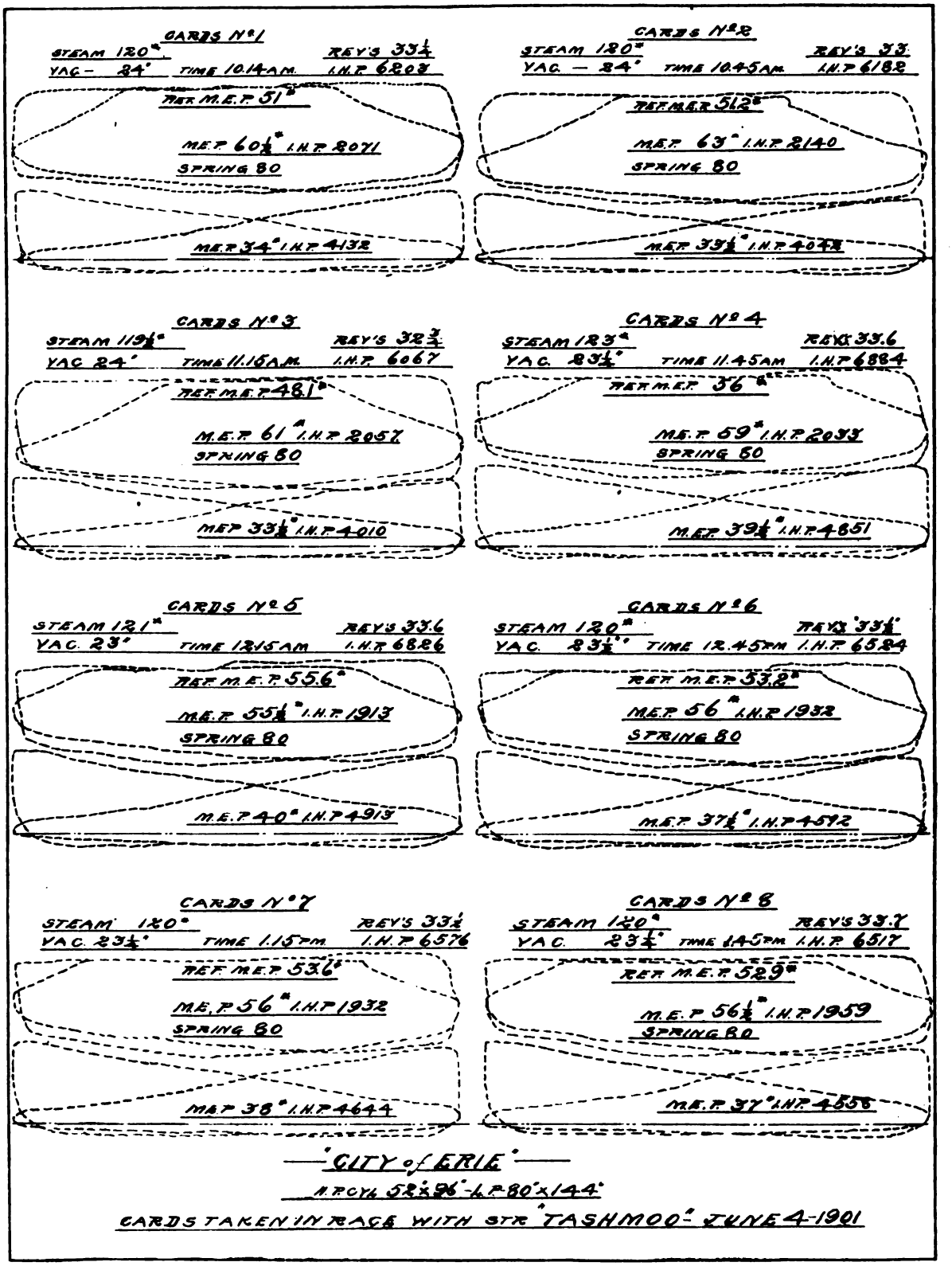

OVERFLOW
INTO PADDLE BOX
OVER FLOW
INTO PADDLE BOX
TANK, CAPACITY 13 TONS OF WATER
TIME REQUIRED TO FILL 7 MINUTES
" " DISCHARGE 45 SECONDS
OUTLET VALVE
OUTLET VA
2½° ANGLE OF HEEL WITH 13 TONS OF WATER IN TANK

To illustrate paper on '
by



STEAMER "C

To illustrate paper on "Trial Speed Between the Steamers City of Erie and Tashmoo," by Frank E. Kirby, Esq., Member of Council.



THE HORTON PETERS CO. PHOTO-LITHO. WASHINGTON, D. C.

To illustrate paper on "Trial of Speed between the Steamers City of Erie and Tashmoo," by Frank E. Kirby, Esq., Member of Council.

To illustrate paper on "Trial of Speed between the Steamers City of Erie and Tashmoo," by Frank E. Kirby, Esq., Member of Council.

To illustrate paper on "The Effect of Variation of Dimensions upon the Stresses in a Ship's Structure," by Prof. Herbert C. Sadler, Member.

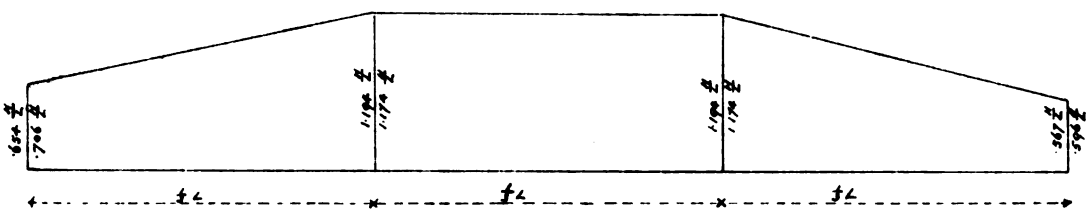
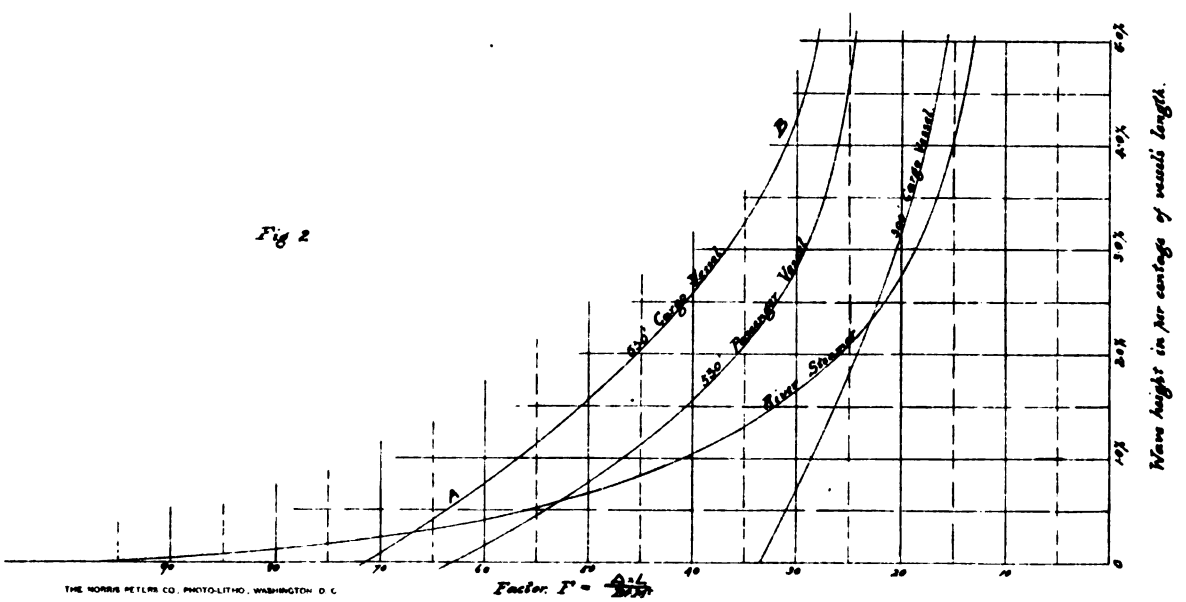


Fig. 4



To illustrate paper on "Power Consumed in Propelling . Speeds," by Frank M. Leavitt, Es

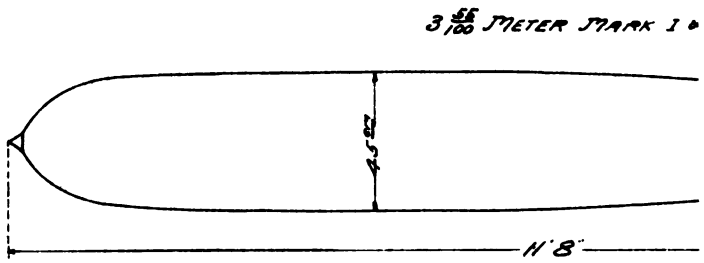


FIG I.

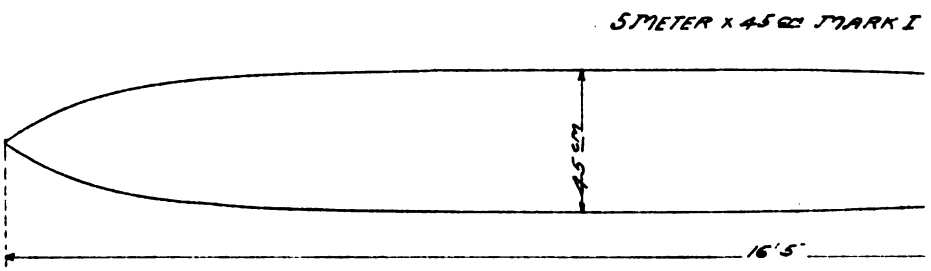
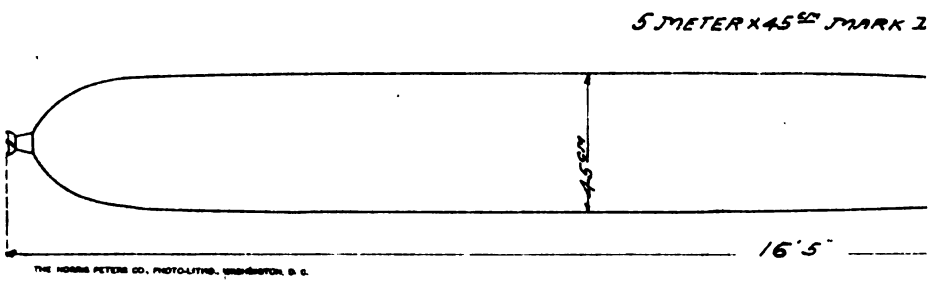


FIG 2

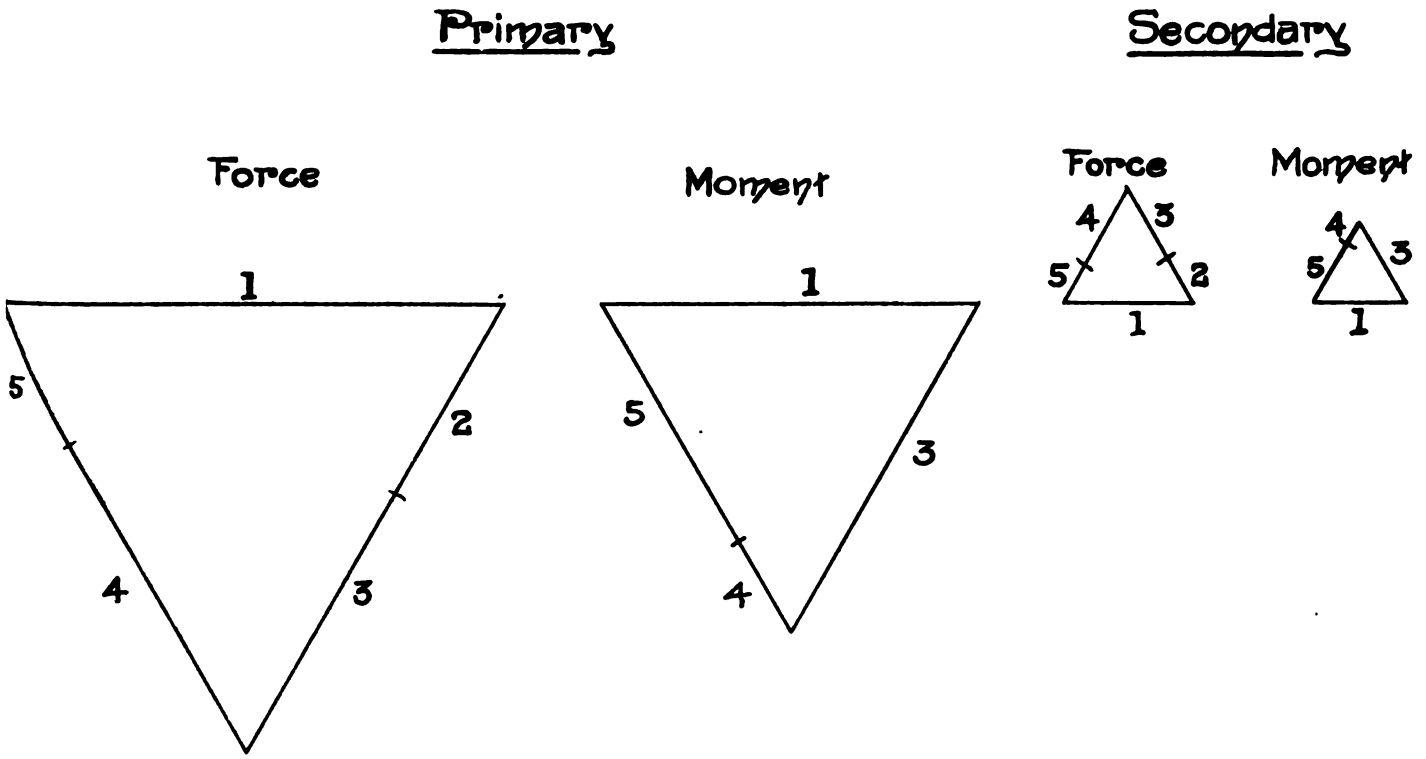


THE HOBBS PETERS CO., PHOTO-LITHO., WASHINGTON, D. C.

FIG 3

To illustrate paper on "The Theoretical and Practical Methods of Balancing Marine Engines," by Naval Constructor D. W. Taylor, U. S. N., Member of Council.

Fig. 29.



Force and Moment Polygons for Fig.

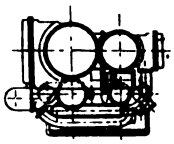
Scales { 1" = 5,000 LBS.
1" = 50,000 LBS. Feet

Moments About No. 2 Crank

To illustrate paper on "The Theoretical and Practical Methods of Balancing Marine Engines," by Rear-Admiral Geo. W. Melville, U. S. N., Vice-President.

BALANCED MARINE ENGINE.

CYLINDERS ..... 27" 39" 56" 80"
STROKE ..... 48"
BOILER PRESSURE ..... 200 LBS.



ARRANGEMENT OF STEAM CONNECTIONS.

Fig. 27.

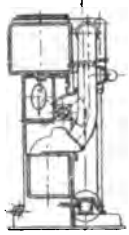


Fig. 25.

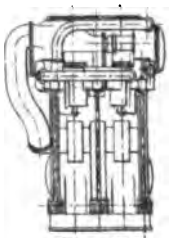


Fig. 26.

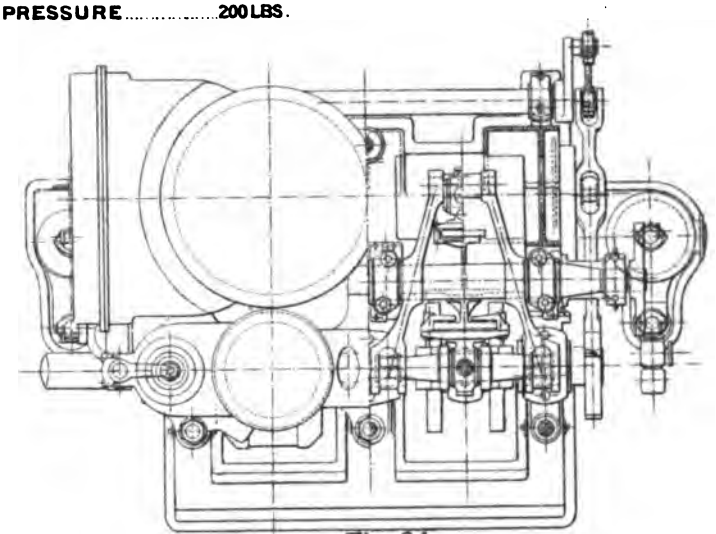


Fig. 24.

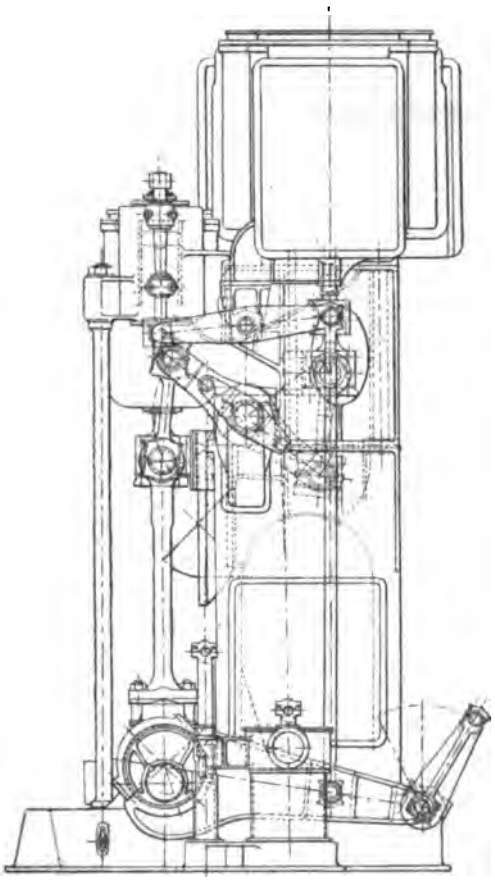


Fig. 22.

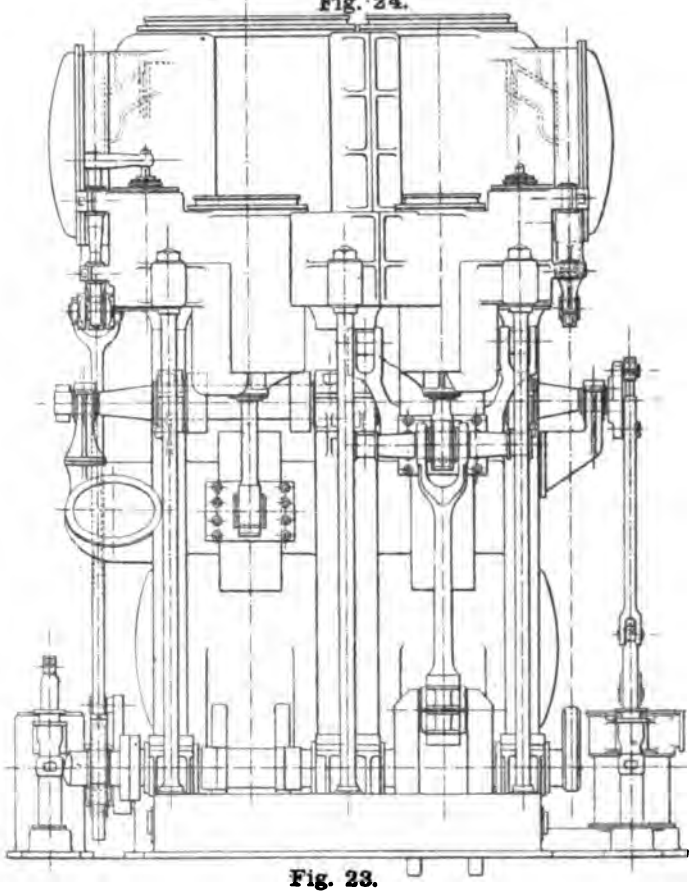


Fig. 23.

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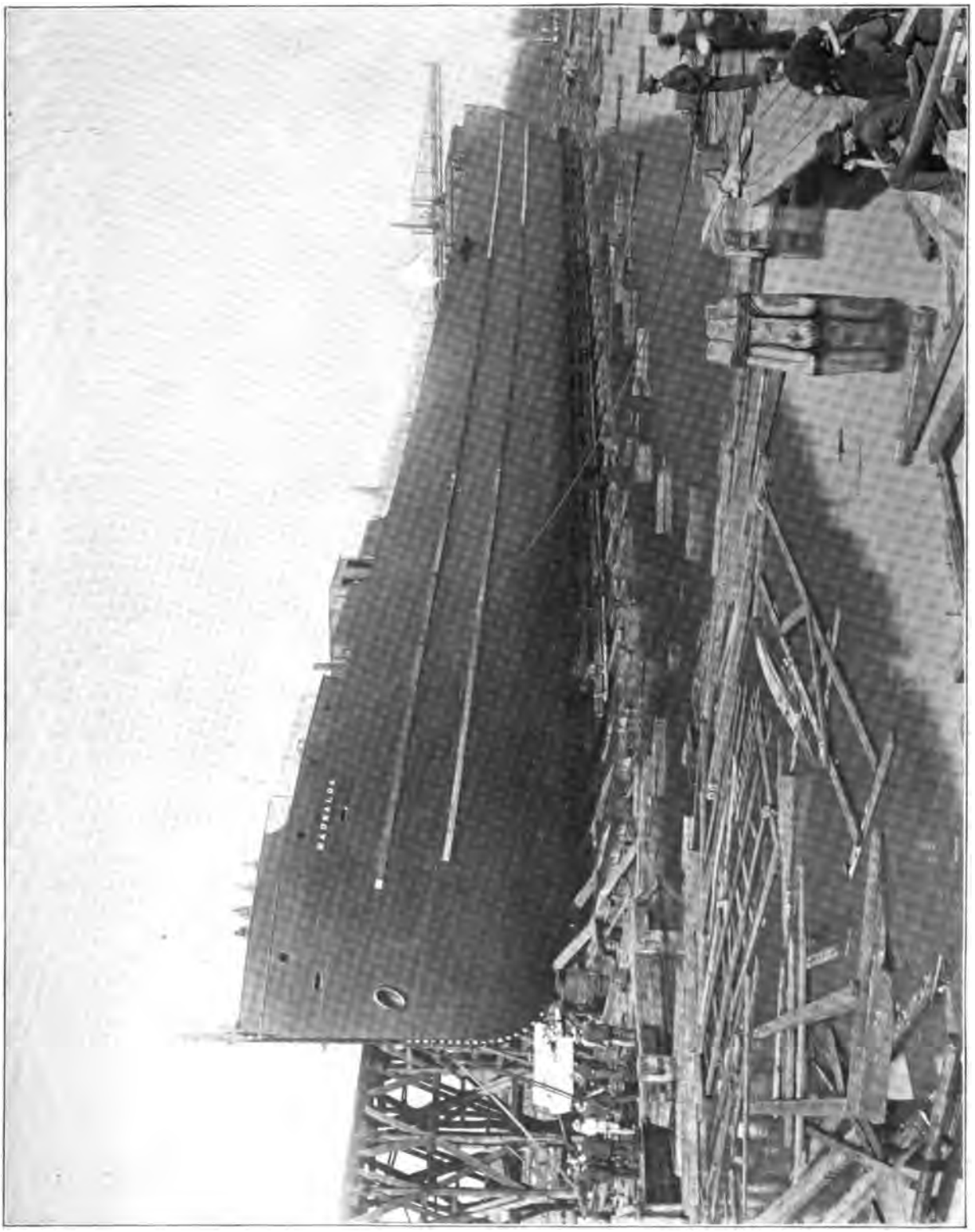

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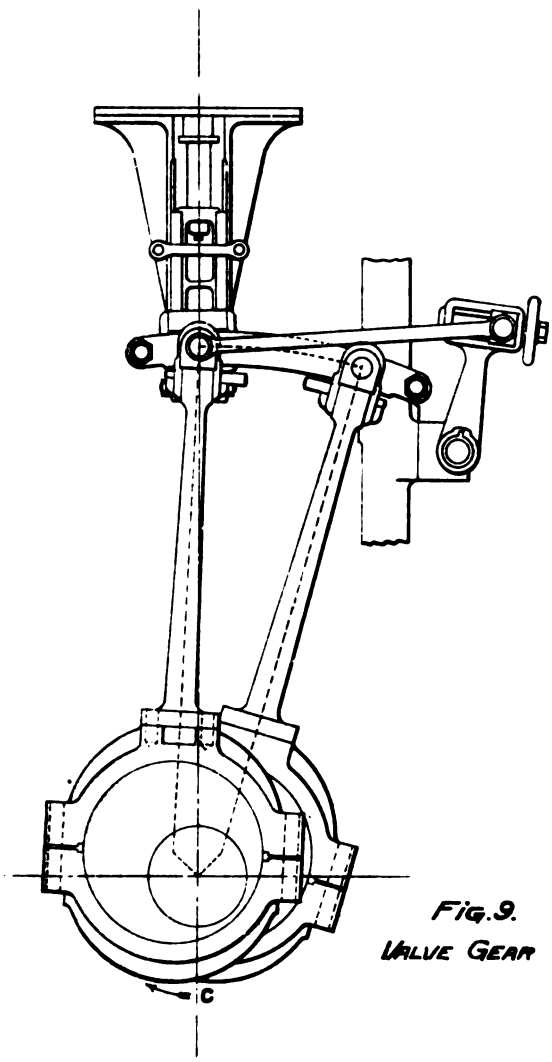

FIG. 9.
VALVE GEAR

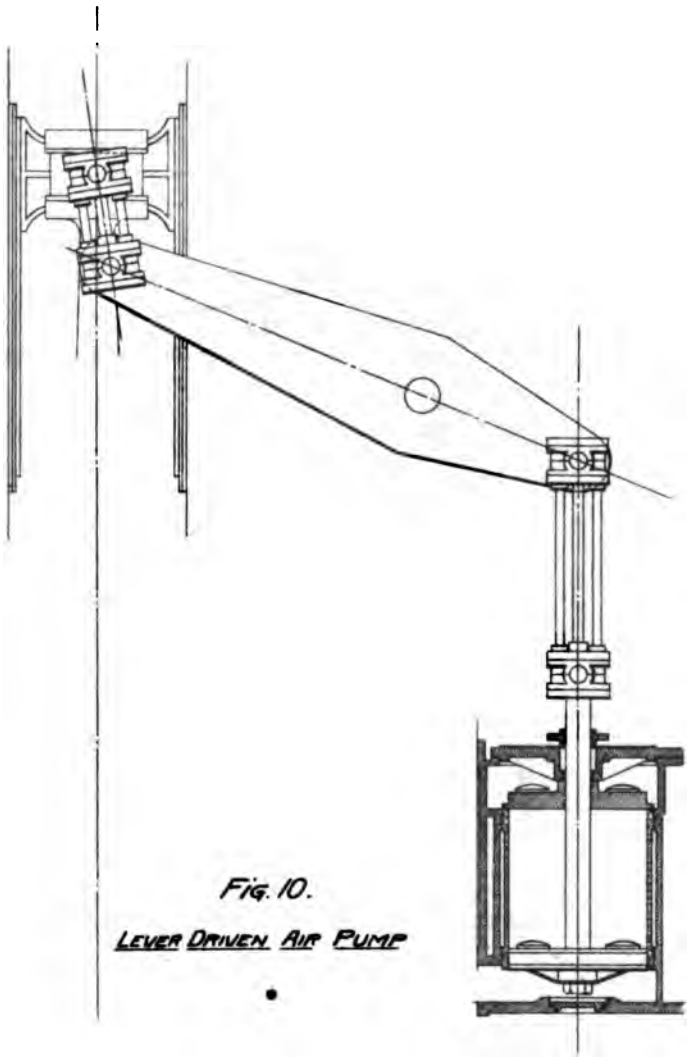

FIG. 10.
LEVER DRIVEN AIR PUMP

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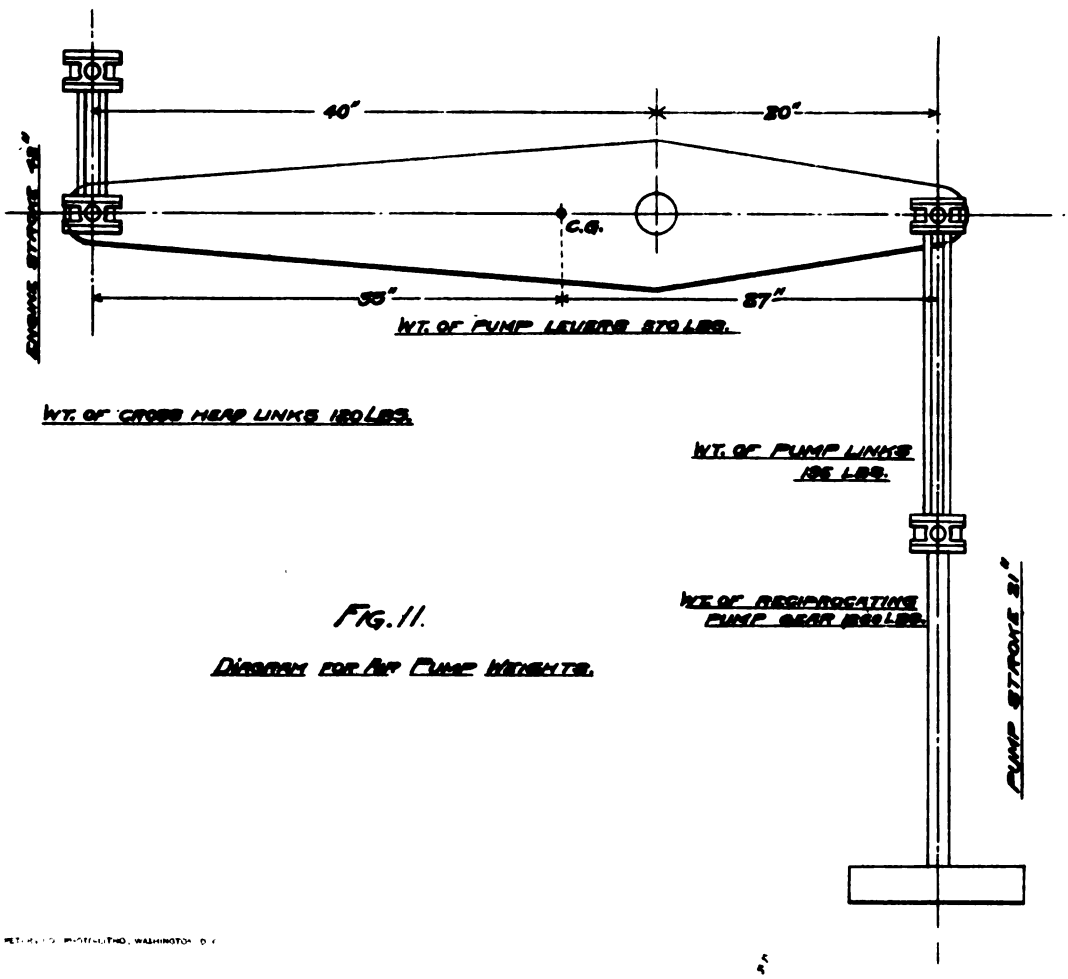

FIG. 11.

DIAGRAM FOR AIR PUMP WEIGHTS.

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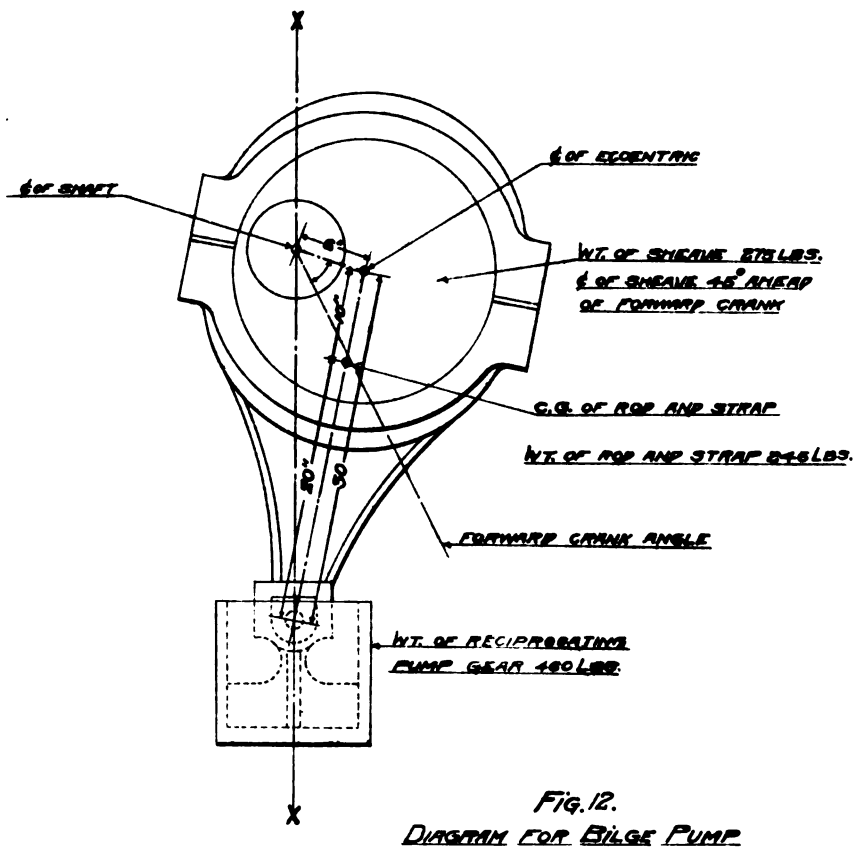
℄ OF SHAFT
℄ OF ECCENTRIC
WT. OF SHEAVE 278 LBS.
℄ OF SHEAVE 45° AHEAD OF FORWARD CRANK
C.G. OF ROD AND STRAP
WT. OF ROD AND STRAP 248 LBS.
FORWARD CRANK ANGLE
WT. OF RECIPROCATING PUMP GEAR 480 LBS.
40"
50"

Fig. 12.
Diagram for Bilge Pump

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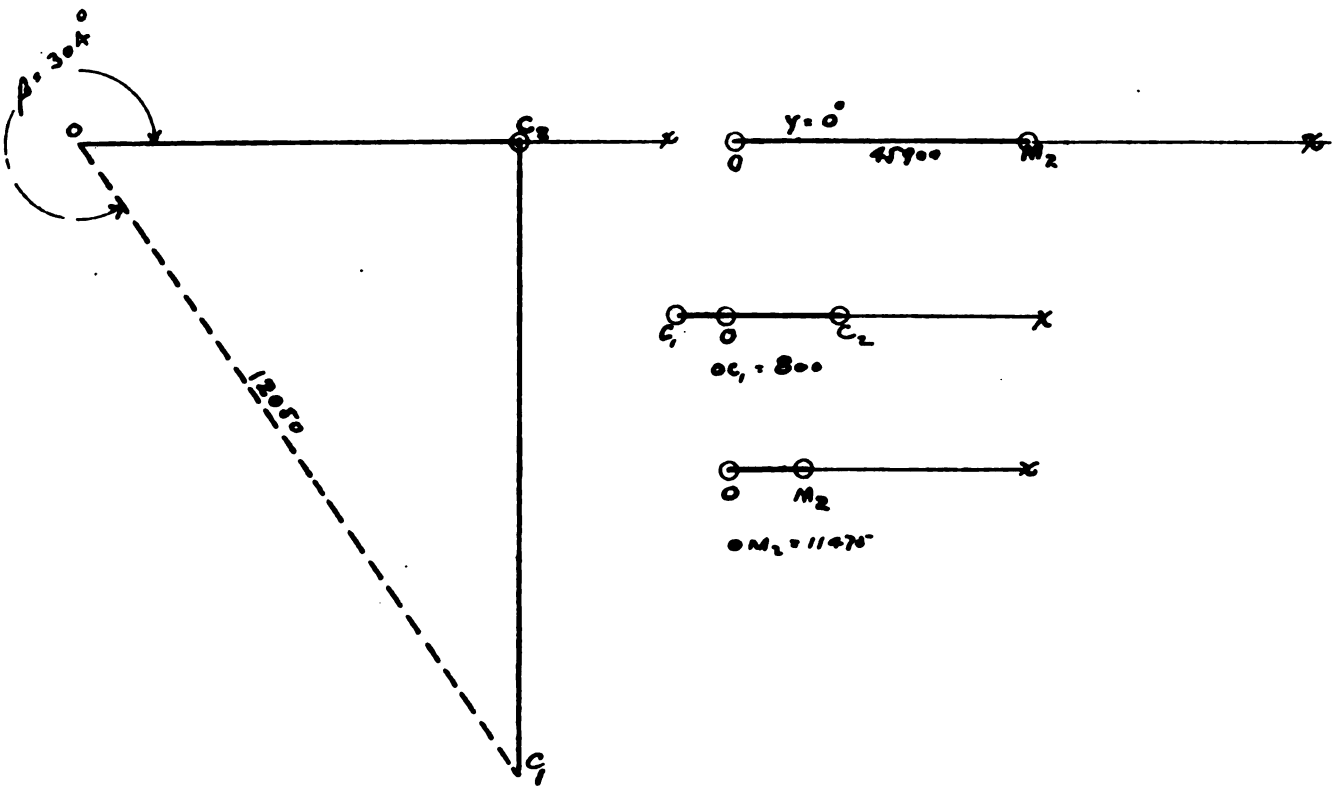

FIG. 13.

CHARLESTON POLYGONS

SCALES

FORCE POLYGONS 1 IN = 3000 LBS

MOMENT do 1 IN = 30000 LB-FT.

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Engines," by Naval Constructor D. W. Taylor, U. S. N

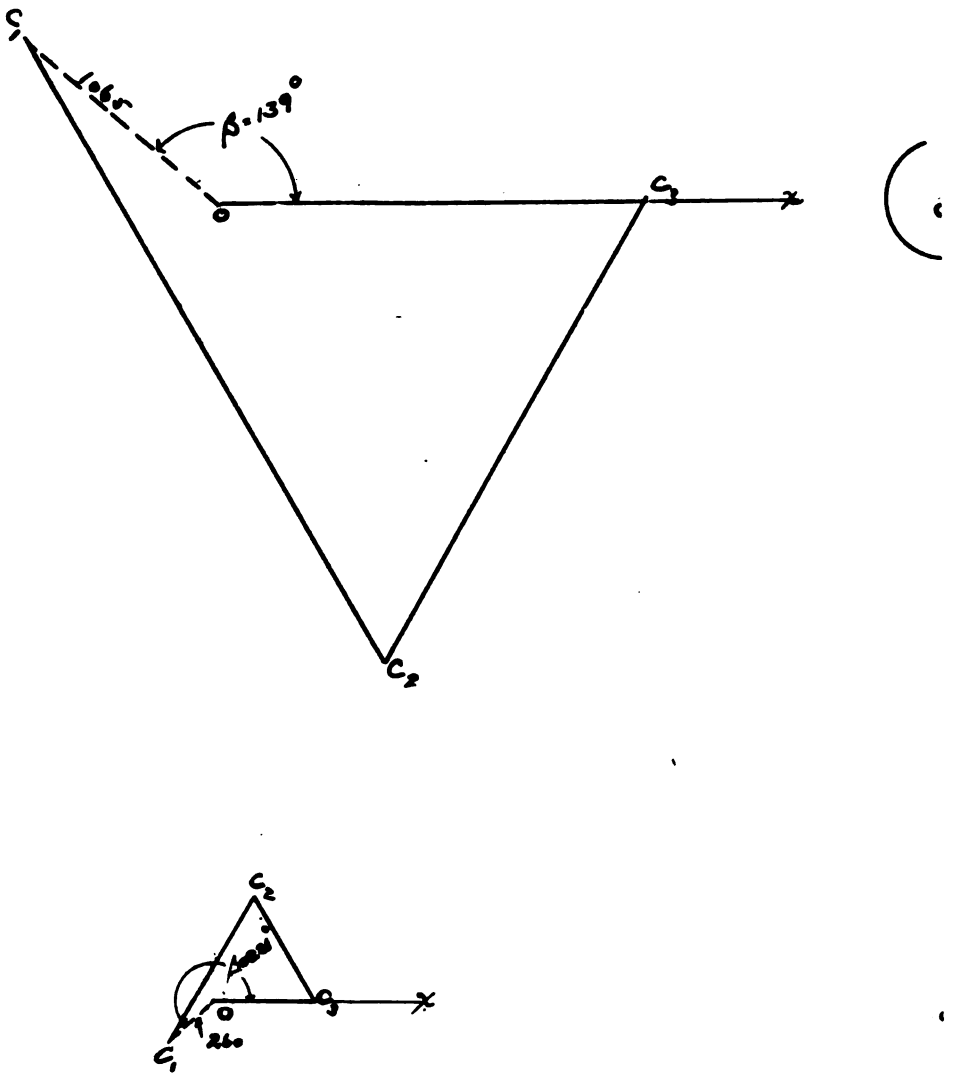

FIG. 14.

YORKTOWN POLYGON

SCALES

FORCE POLYGONS 1 IN = 800 L

MOMENT do 1 IN = 8000 L

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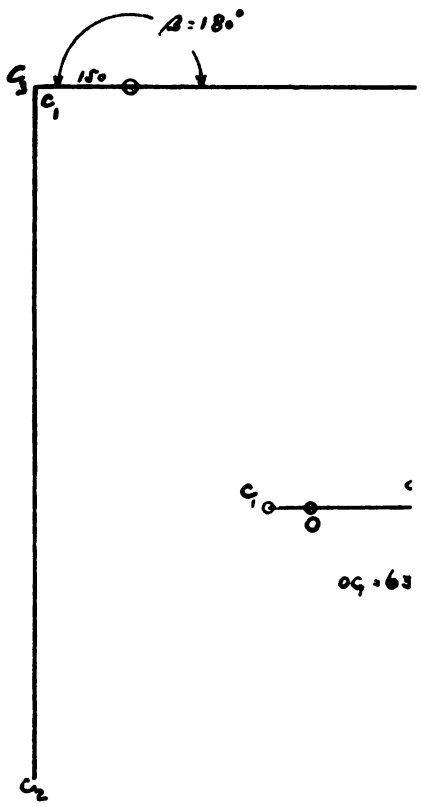
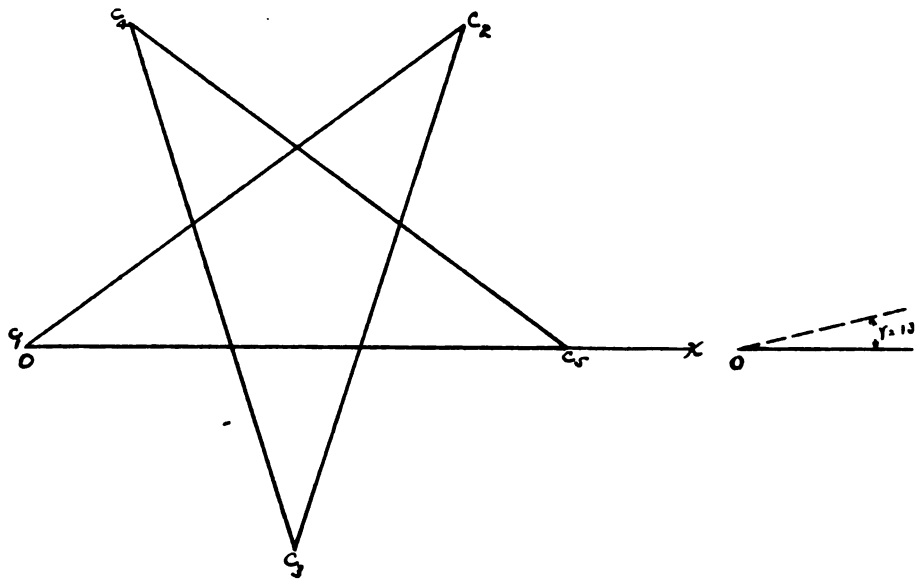

β = 180°
150
C_3
C_1
C_2
C_1
C
O
$OC_1 = 63$

To illustrate paper on "The Theoretical and Practical Method. Engines," by Naval Constructor D. W. Taylor, U. S. N.,



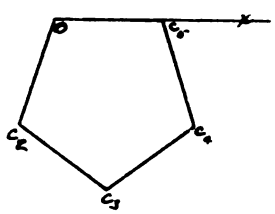


FIG. 16.

CUSHING POLYGONS

SCALES

FORCE POLYGONS 1 IN. = 50 LBS

MOMENT do 1 IN. = 2000 LB

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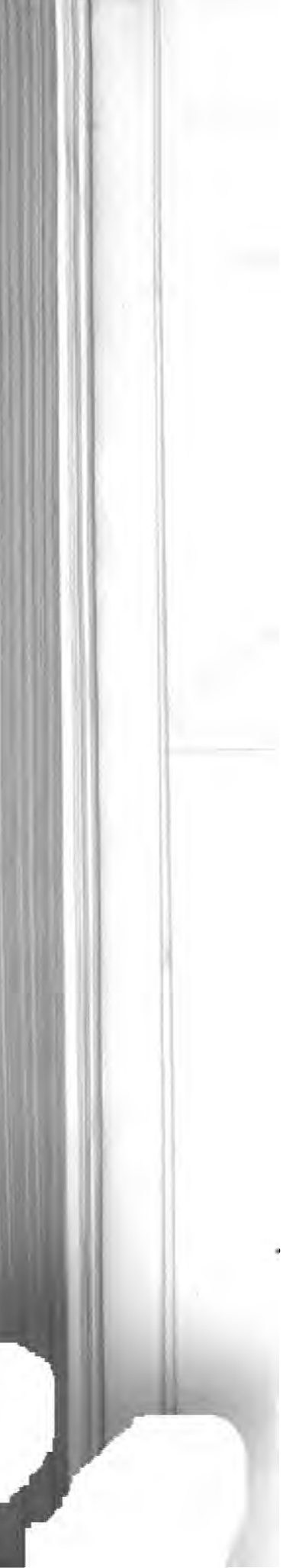

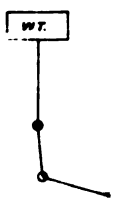

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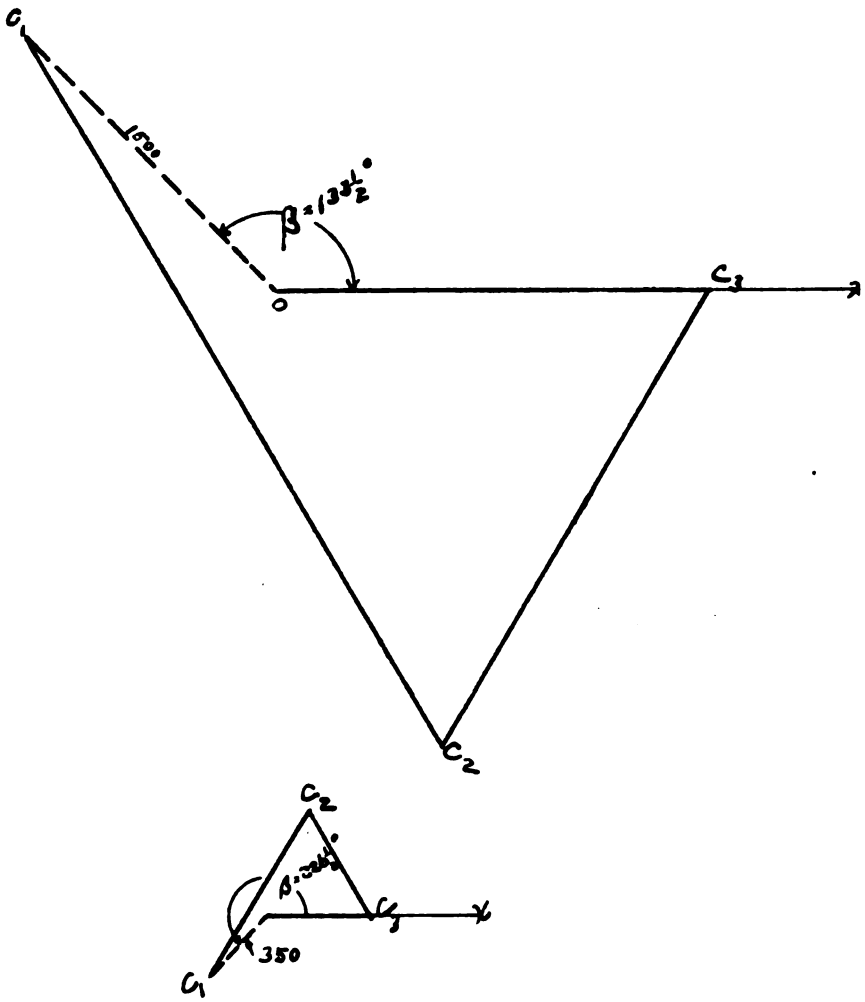

FIG. 23.

YORKTOWN

SCALES

FORCE POLYGONS 1/.

MOMENT do 1/

NOTE! WEIGHT OF LOW PRESSUR
INCREASED TO

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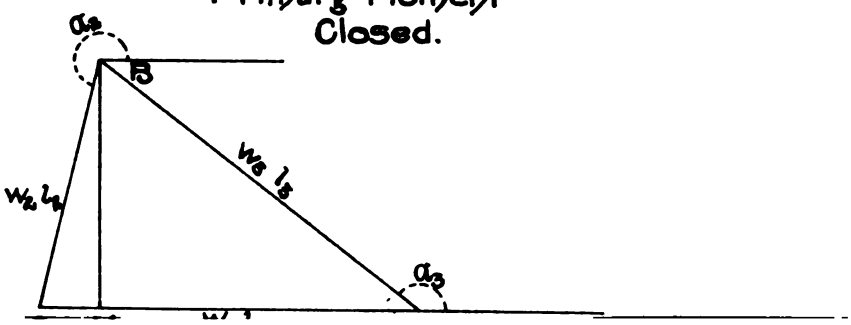
Fig. 24.
Primary Moment
Closed.



NTS FOR 4 CRANK CONNECTING ROD

	'75			
2	'5	1	15	2

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Engines," by Naval Constructor D. W. Taylor, U. S.

Fig. 26.

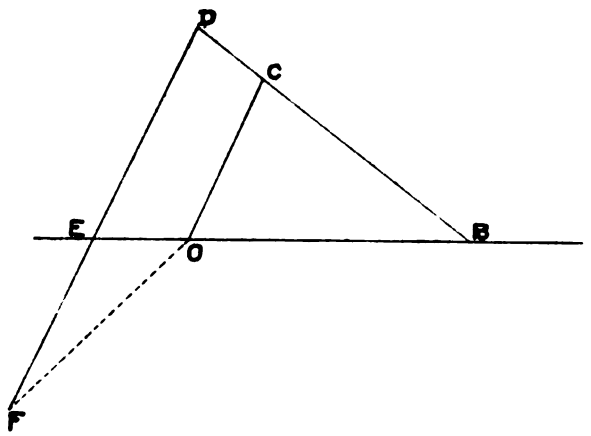

Fig. 28.

COMPLETELY BALANCED
FIVE CRANK ARRANGEMENT.

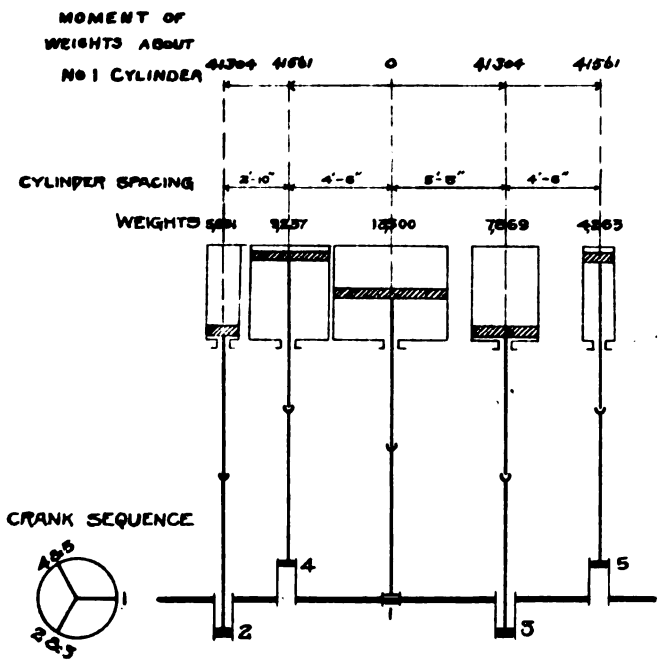

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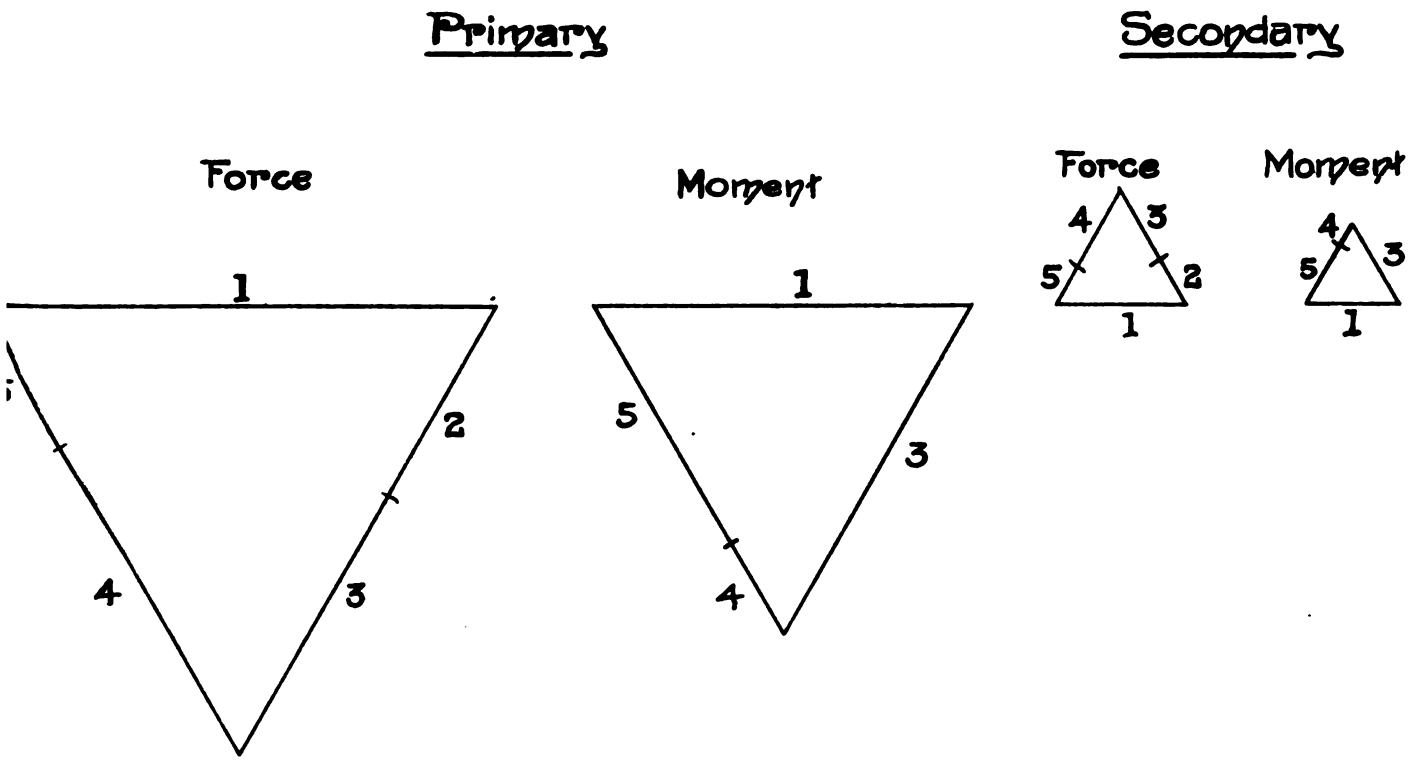
Fig. 29.

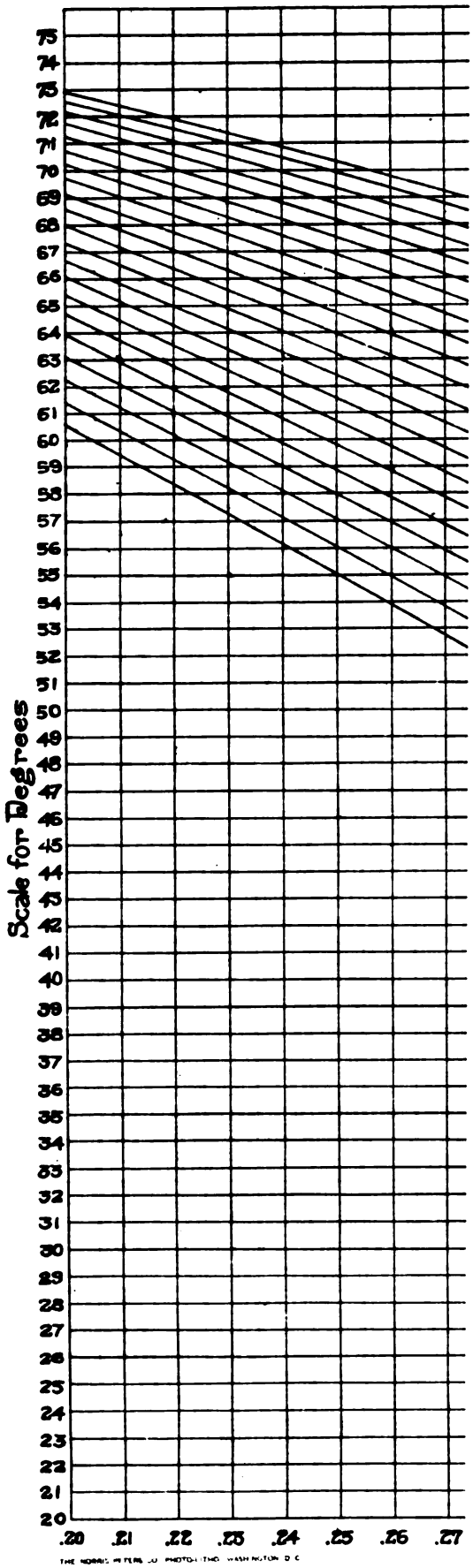


Force and Moment Polygons for Fig.

Scales { 1" = 5,000 LBS.
1" = 50,000 LBS. Feet

Moments About No. 2 Crank

Diagram A.


Scale for Degrees
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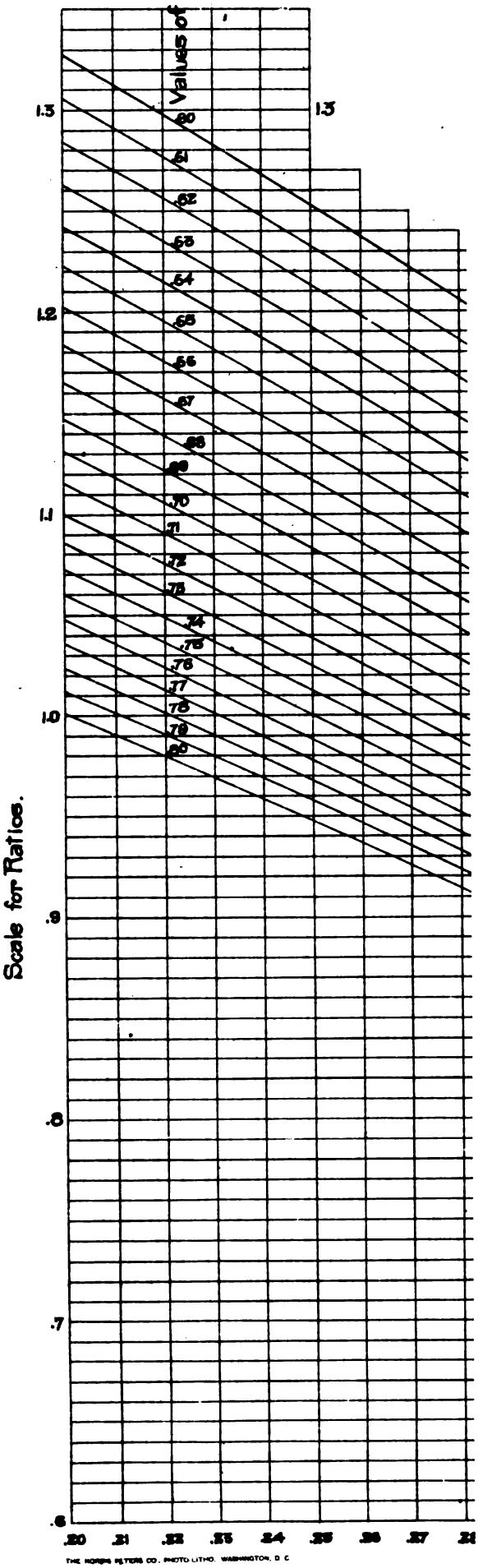

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To illustra
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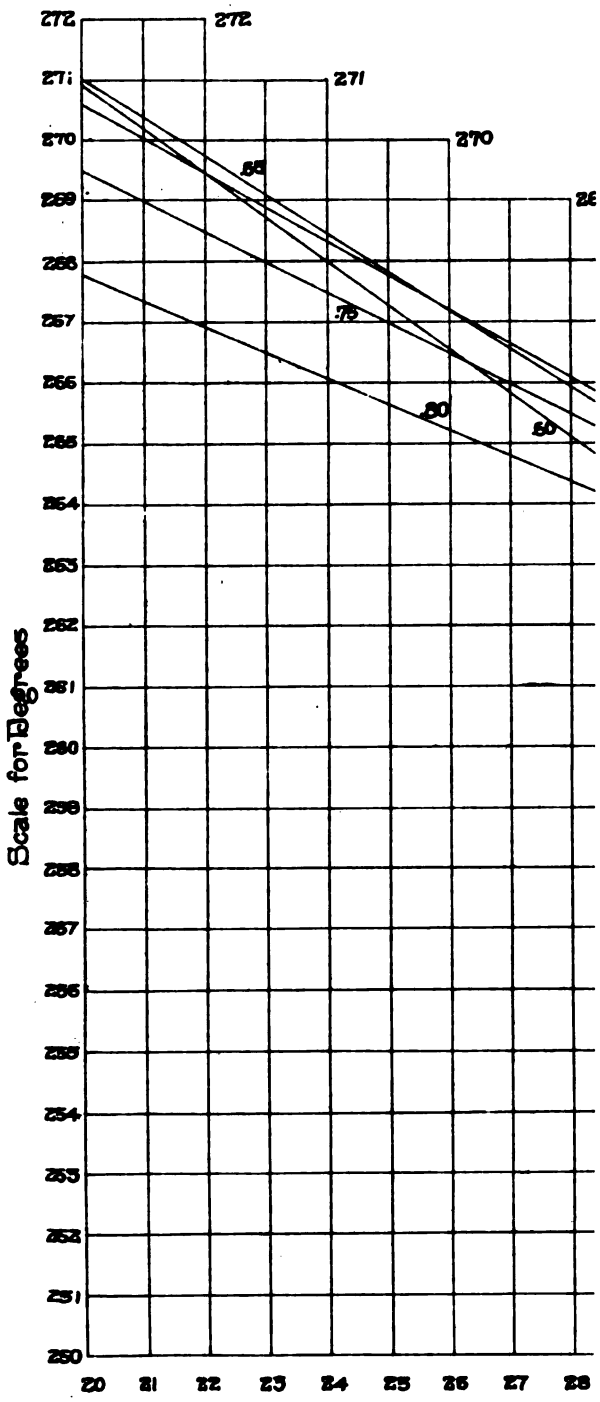

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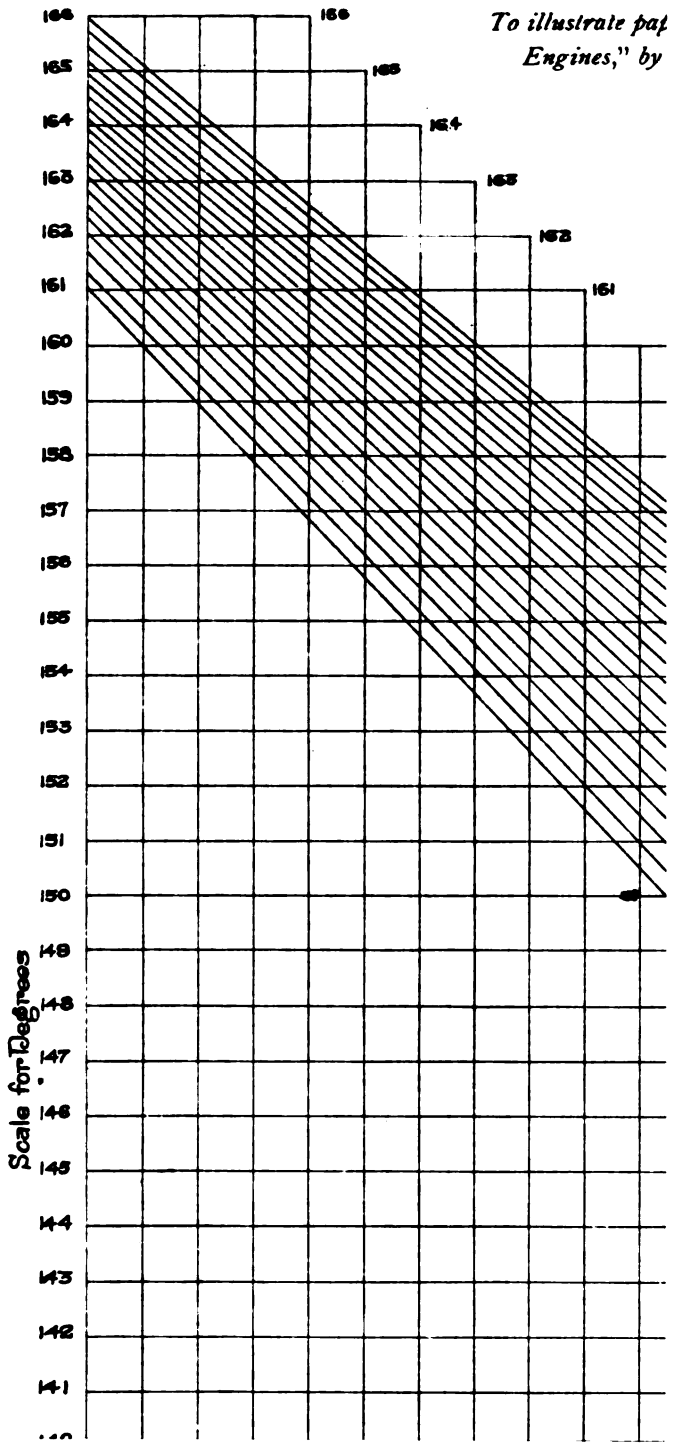

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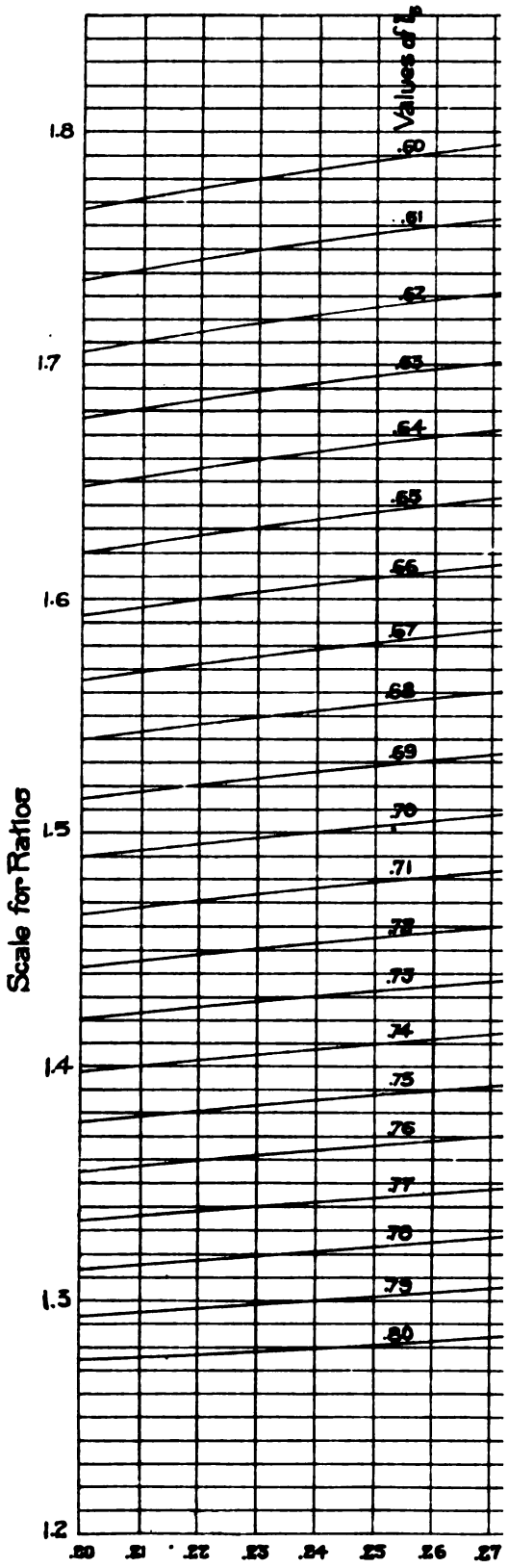
Values of L_p
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.62
.63
.64
.65
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.67
.68
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.72
.73
.74
.75
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.80
Scale for Ratios
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1.7
1.6
1.5
1.4
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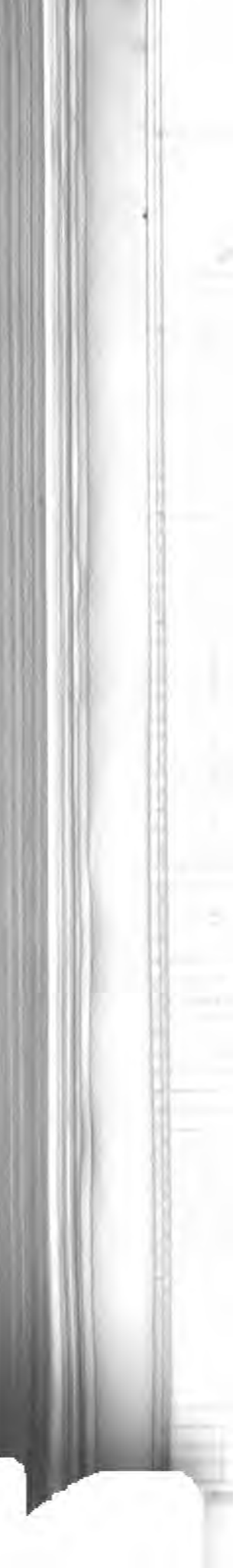

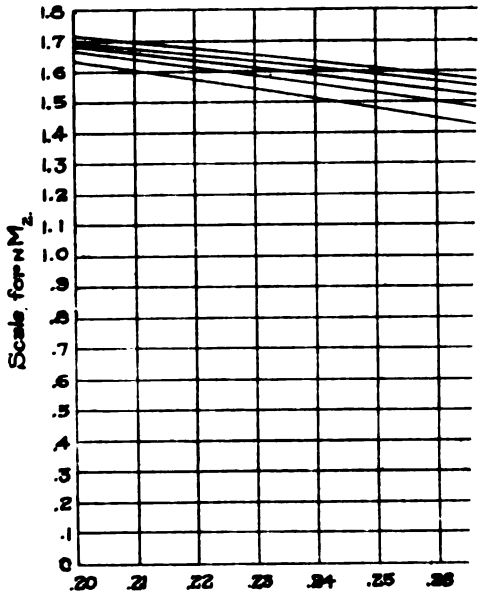

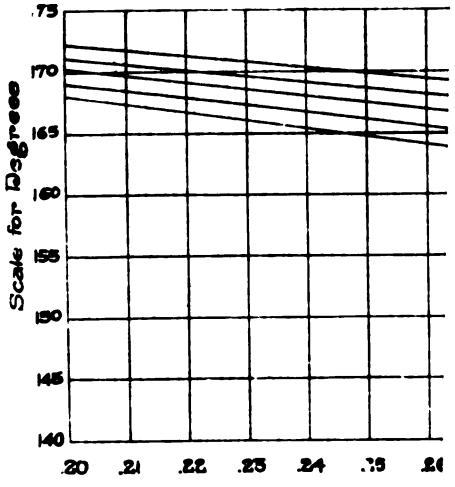

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B A_

L.P. Piston
Fig. 15.

1st I.P. Pis. 2d
Fig. 14.

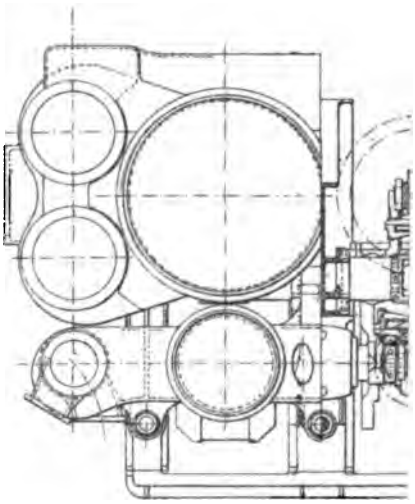

Fig. 9.

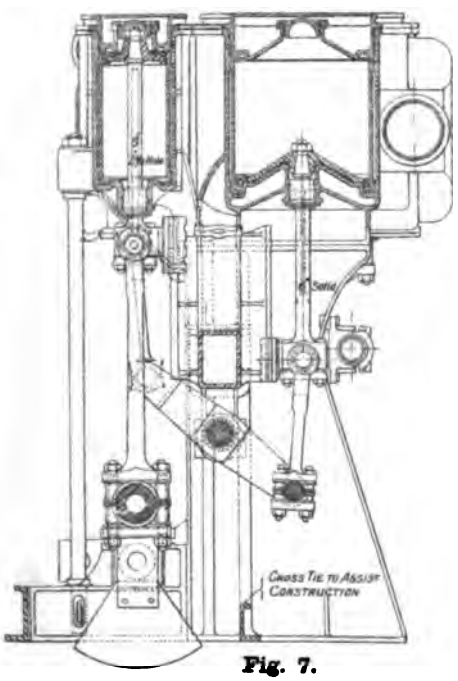

Fig. 7.

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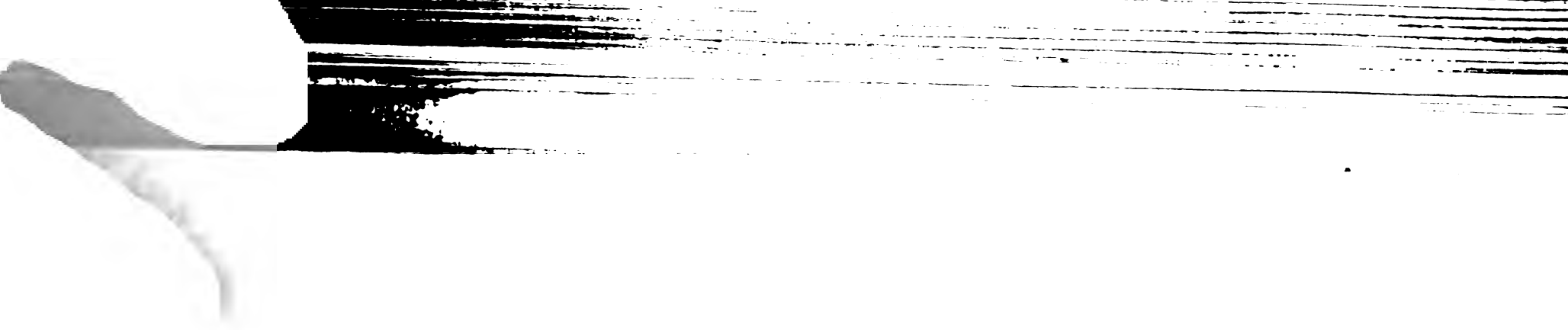
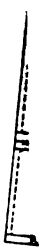
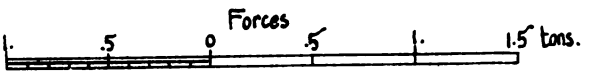


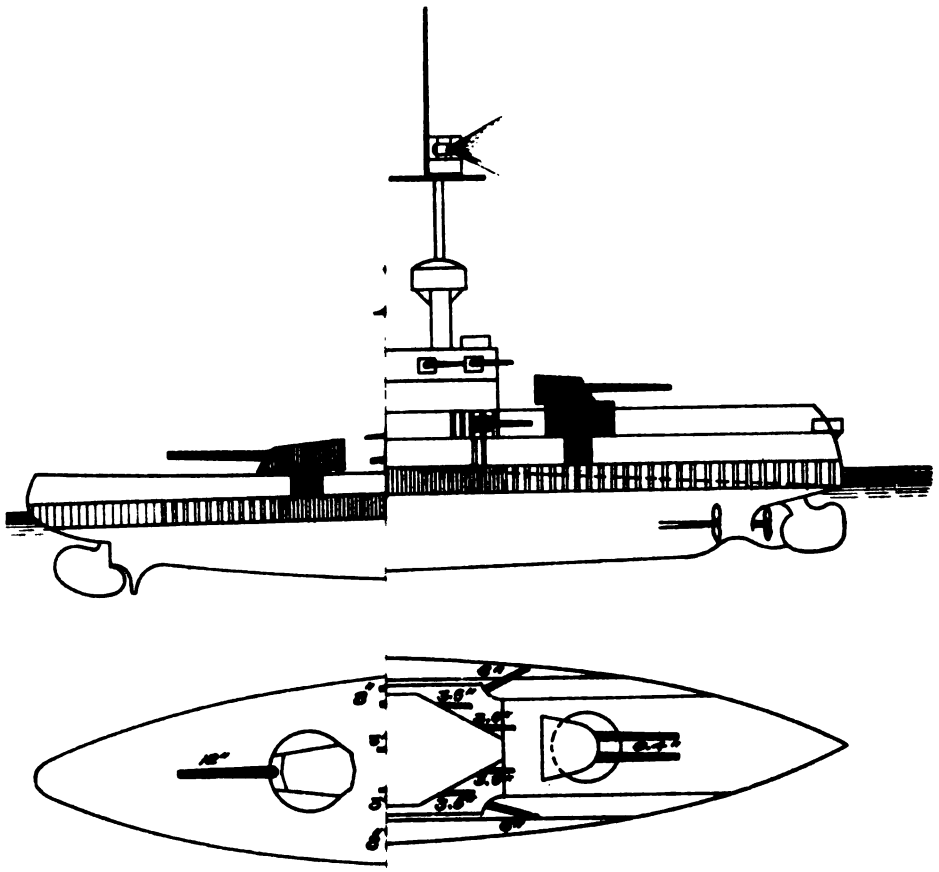


Tra

ırrow-Schlick-Tweedy System.

ıoments of 2nd and 4th periods.





VITTORIO EMANUE KNOTS, 11,800 TONS

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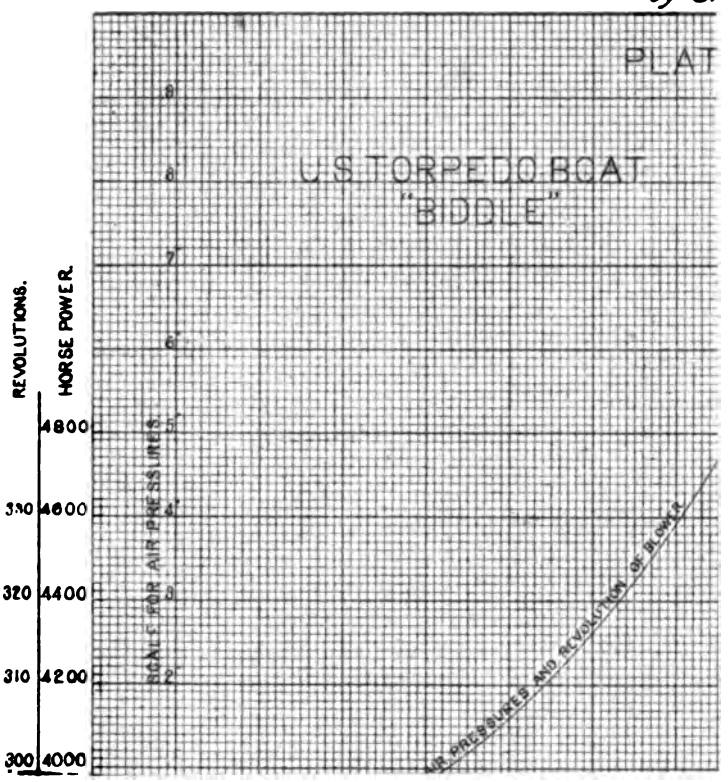
by C



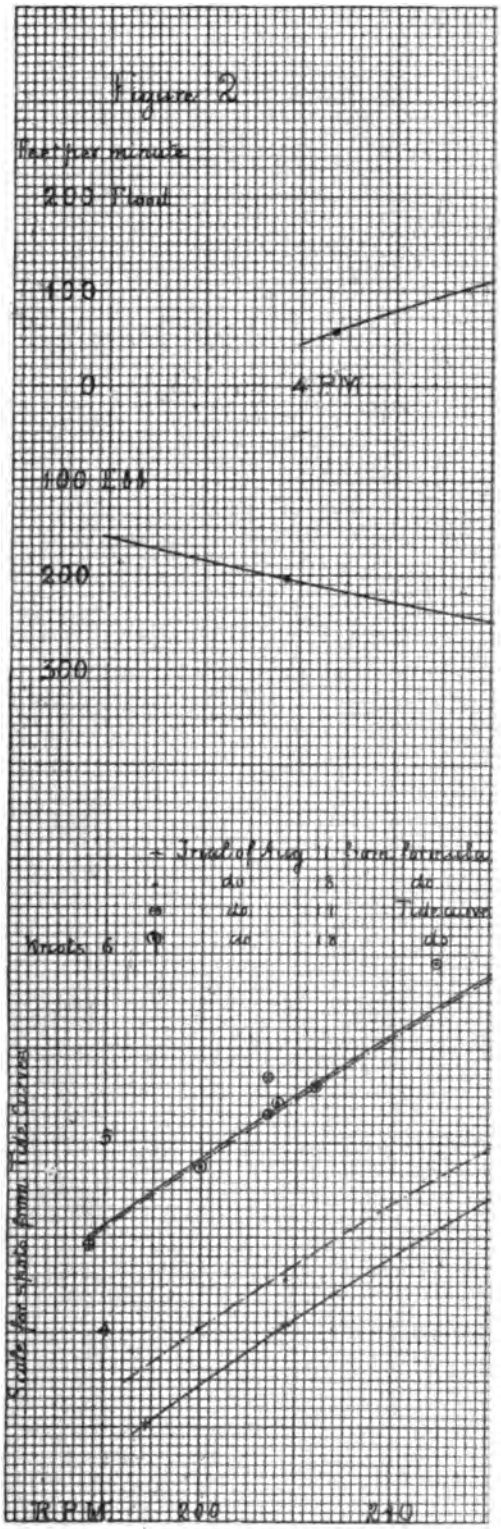
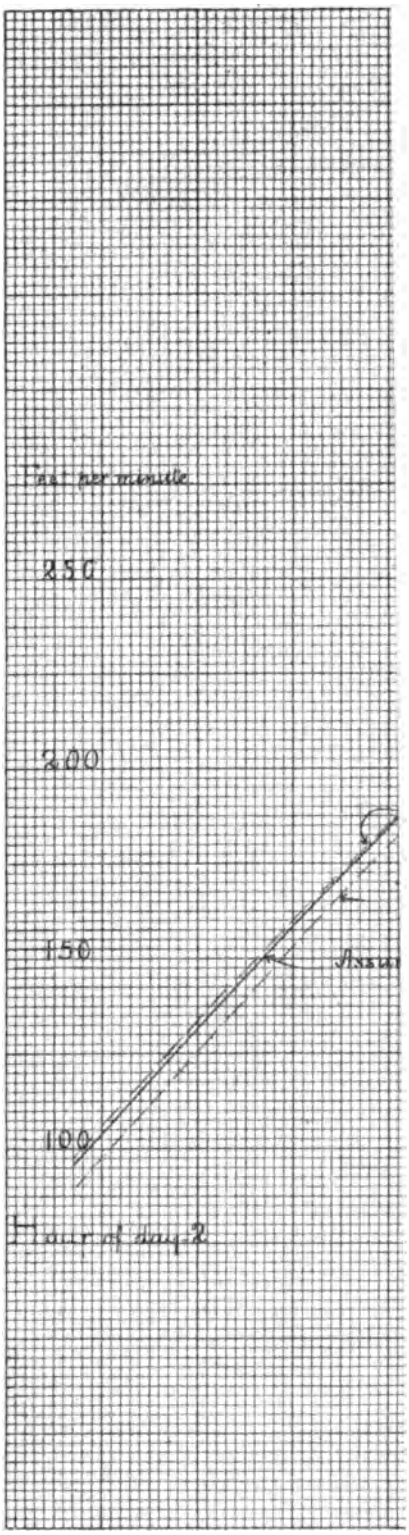
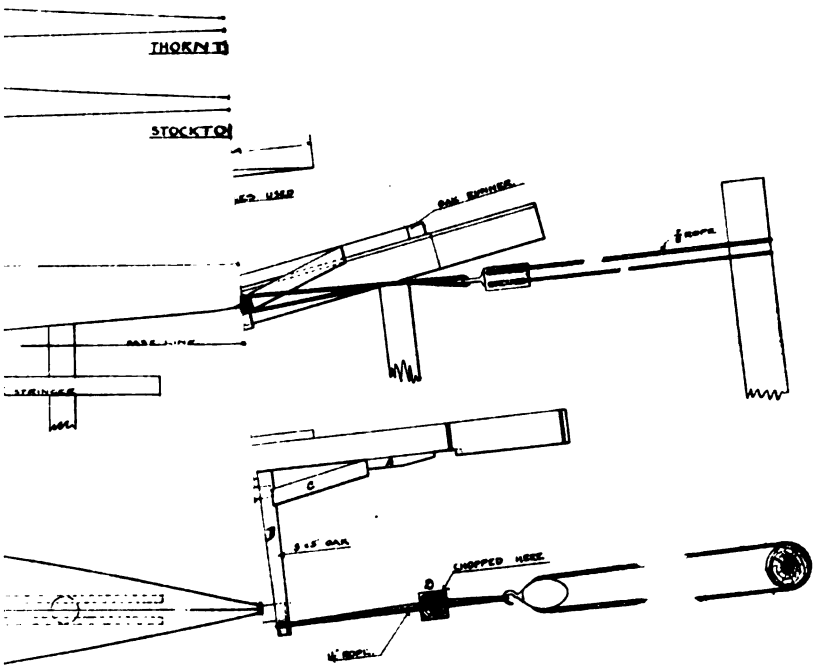


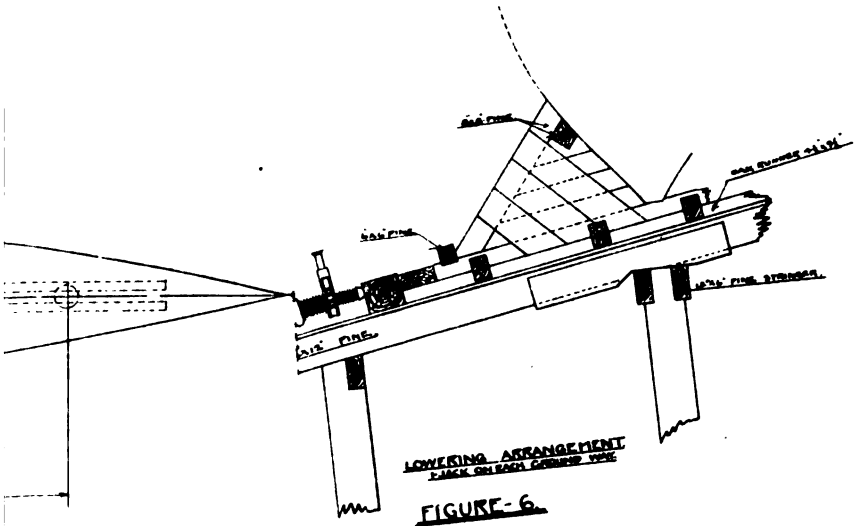
Figure 2
Feet per minute
200 Fland
100
0
4 PM
100 Ebb
200
300
Knots 6
RPM
200
210


Feet per minute
250
200
150
100
Hour of day

To illustrate pap
by Ass



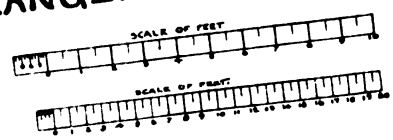
L ARRANGEMENT.
IRE-5.



LOWERING ARRANGEMENT
FIGURE-6.

TORPEDO BOATS

LANGEMENT OF LAUNCH.



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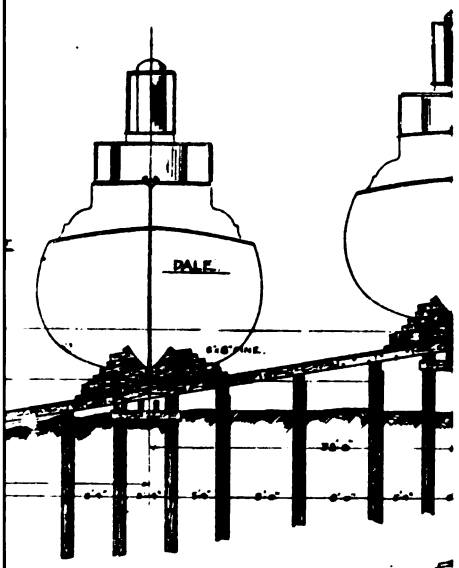
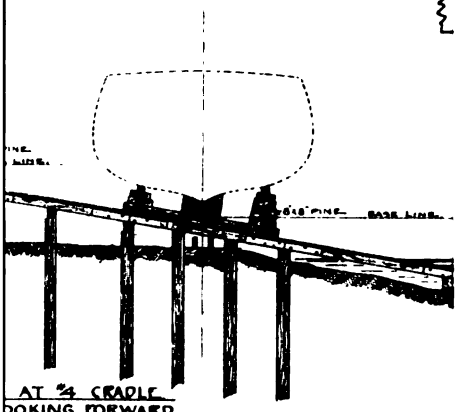


BOW VIEW LOOKING AFT.

FIGURE-3.



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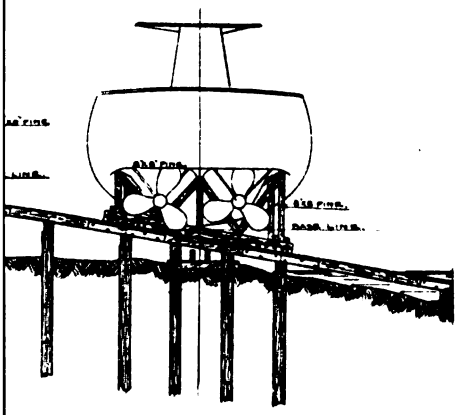
OOKING FORWARD.

FIGURE-4.



VIEW LOOKING FORWARD.

FIGURE-5.

To illustrate paper on "Side Launch of Torpedo-Boats and Torpedo-Boat Destroyers," by Asst. Naval Constructor Wm. G. Groesbeck, U. S. N., Member.

To illustrate paper on "Side Launch of Torpedo-Boats and Torpedo-Boat Destroyers," by Asst. Naval Constructor Wm. G. Groesbeck, U. S. N., Member.

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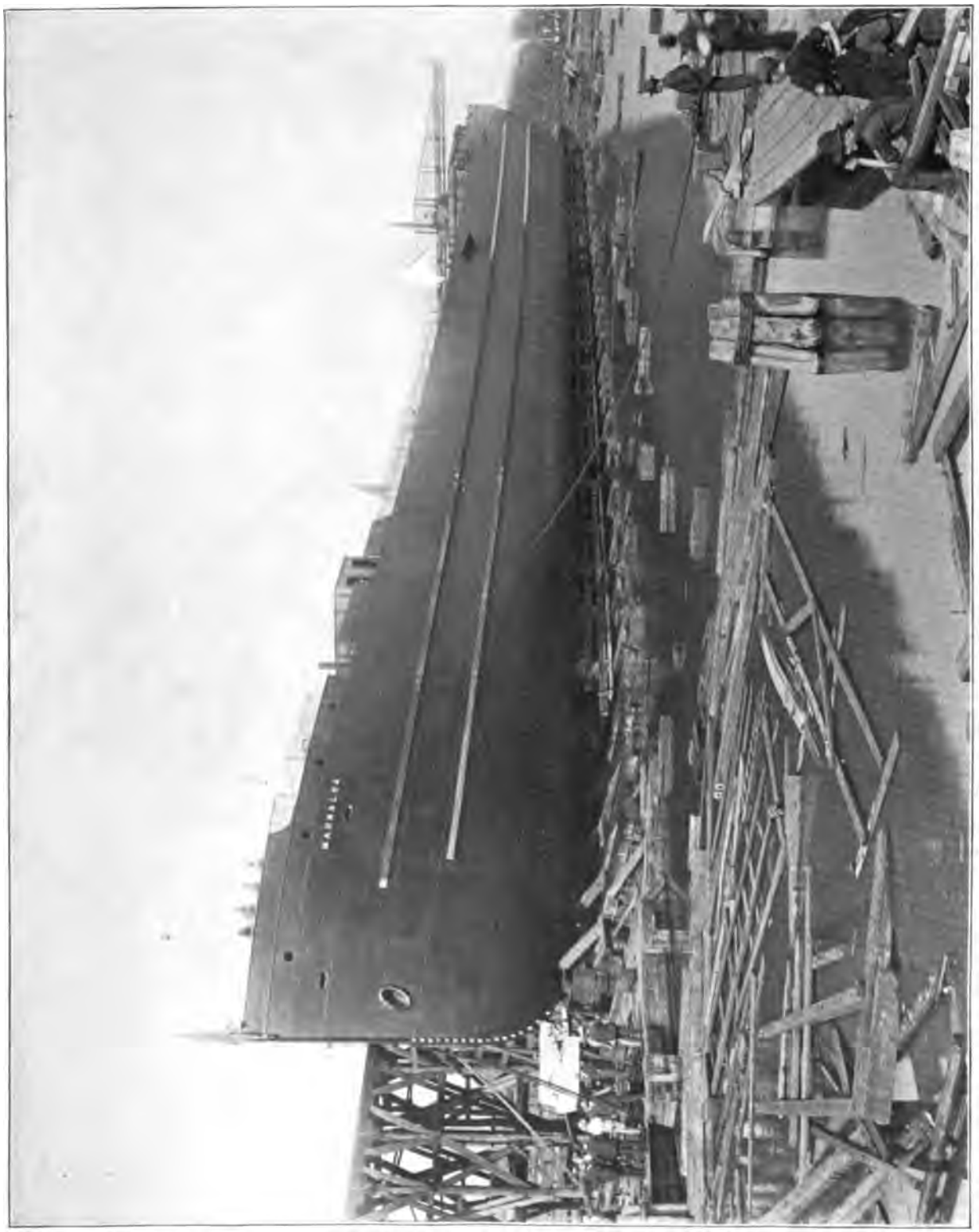

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